SOME ASPECTS OF MODELLING AND CONTROL OF AUTOMOTIVE POWER SYSTEMS

by

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To my wife Elizabeth, in appreciation of her continuous support and encouragement.

CONTENTS

CHAPTER 3

BACKGROUND TO AUTOMOTIVE SYSTEM DEVELOPMENT AND MODELLING

CHAPTER 4

VEHICLE POWERTRAIN SYSTEMS SIMULATION

LIST OF TABLES

LIST OF FIGURES

CHAPTER 6 Page

CHAPTER 7 Page

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I should like to thank the Ford Motor Company Ltd., for providing certain engine data for use in this research; without the availability of such data it would have been difficult to complete much of the work. Thanks are due also to Mr. D.B. Hodgson of the Lucas Research Centre for his willingness to support the work.

The author is grateful to his supervisor Dr. M.T.G. Hughes for his recommendations and encouragement; to Mr. A.J. Hulme for his assistance in operating the Prime 550 computer; and also to Dr. R.P. Jones of the Department of Engineering for the general advice which he has given.

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SUMMARY

With the increasing development of new vehicles with advanced powertrains, and the increasing prominence of energy and environmental considerations, sophisticated computer aided analysis and design tools are rapidly becoming essential. Computer aided engineering (CAE) in automotive applications has gradually extended its boundaries beyond geometrical modelling, finite element techniques, and drafting, into automotive engine management and driveline control. In this particular area of activity the CAE techniques proving most useful are static and dynamic systems modelling and simulation, and control systems synthesis and optimisation.

The main emphasis in this thesis is in the area of modelling and simulation of vehicles for addressing problems of emission constrained fuel economy optimisation. In order to place engine control system optimisation in context an assessment of the development potential of automotive systems, with regard to fuel economy improvements, is undertaken.

Some important aspects of the modelling of automotive systems are addressed, with particular relevance to emission constrained fuel economy problems. A dynamic simulation facility, based on this modelling, is developed for use in the preliminary evaluation of control strategies. The thesis reports on an application of this facility to the study of an engine calibration procedure intended for on-line optimisation of the dynamic system; and also to the study of a simple extremum-seeking adaptive ignition timing controller. The simulation facility proved useful in the study of these two different types of control procedures, but more importantly, served as a means of identifying the more general requirements associated with the use of continuous dynamic simulation in the context of automotive control system development.

Finally, recommendations are made for the development of an integrated CAE environment for use by automotive systems engineers. These recommendations are based on the diverse requirements of the data handling, modelling, and simulation activities encountered during the course of this study.

CHAPTER 1

INTRODUCTION

1.1 COMPUTER AIDED ENGINEERING.

Computer aided engineering (CAE), which term encompasses computer aided design or drafting (CAD), is now an integral part of the modern automotive industry. Geometrical modelling and finite element techniques, frequently used for such tasks as stress analysis and weight optimisation, have more recently been complemented by techniques applicable to the design and development of control systems for powertrains.

In the area of automotive engine management and driveline control systems the CAE techniques proving most useful are dynamic systems modelling and simulation, and control systems synthesis and optimisation. Dynamic system simulation is an analysis tool intended for use early in the design process; it allows evaluation of concepts, comparison of alternatives, control strategy development, and sensitivity studies to be performed; as well as the tuning of parameters and the selection and matching of engine and transmission components. With the development of new and advanced vehicle powertrains and the increased awareness of the need for energy and environmental conservation, these CAE tools are rapidly becoming essential.

1.2 OBJECTIVES OF THE RESEARCH WORK.

The main emphasis of this work is in the modelling and simulation aspects of vehicles for fuel economy studies; encompassing, in particular, the field of engine calibration and control.

There are many aspects of automobile design and control that have a bearing on fuel economy and emissions (Table 1.1). It will be realised that in order to develop vehicles that have improved performance with regard to these factors, each component of the system cannot be viewed in isolation. Automotive systems exhibit complex interactions and present formidable problems in identification, assessment, and optimisation.

In order to place engine control problems in their proper context a review and assessment of the development potential of automotive systems had to be made. This was a necessary precedence to the development of a simulation model which was a vital part of the work. The model enables rapid prediction and evaluation of the effects of parameter changes, and is used specifically to study certain optimal and adaptive control strategies.

The main part of the work involves systems employing manually shifted stepped ratio transmissions and spark ignition engines as have almost universal application in the saloon car population of Europe. The systems approach, and modelling techniques presented here do, of course, have wide application in the study of all types of automotive vehicles.

TABLE 1.1

Main factors influencing automotive fuel consumption.

1. The Vehicle.

   Engine design and control.
   Transmission and drive-train characteristics.
   Tyre size, type and condition.
   Vehicle size, weight and aerodynamics.
   Fuel characteristics.
   Manufacturing tolerances.
   Maintenance (automotive system condition).

2. Driver Behaviour.

   Velocity change rates.
   Cruising speed.
   Number and timing of gear changes.
   Number of corners and cornering speed.
   Use of brakes.
   Characteristics of journey.
   Proportion of 'cold' operation.

3. Road Conditions.

   Road geometry (camber, radius of curves).
   Road gradient.
   Surface characteristics.
   Speed limits and traffic controls.
   Traffic volume and composition.

4. Environment.

   Barometric pressure temperature and humidity.
   Wind speed and direction.
   Rain, snow and ice effects on road surface.

1.3 ORGANISATION OF THE THESIS.

Chapter 2 introduces the topic of engine control systems optimisation, and presents the problem of emission and driveability constrained fuel economy; providing also an assessment of some current approaches to the solution of such problems.

Background considerations, pertinent to modelling of automotive systems for fuel economy work, are provided by Chapter 3; which also indicates the scope of development, and the author's interpretation of the development potential, of automotive systems in this regard. The chapter also briefly describes a number of fundamental engine characteristics as a priori information, when the models are to be used in addressing emission constrained fuel economy optimisation problems.

Many aspects of the modelling of road vehicles are addressed in Chapter 4. The automotive system is considered in terms of five main interacting subsystems, viz. vehicle, engine, transmission, controller, and driver. While some aspects of the modelling of the vehicle and transmission may be largely familiar, a novel approach is taken to the incorporation of certain dynamic effects of temperature on emissions and fuelling. A fresh approach is also taken to the modelling of driver functions, specifically in regard to vehicles equipped with manually-shifted stepped-ratio transmissions used in a driving schedule tracking role.

Chapter 5 describes an application of the automotive system modelling presented in the previous chapter. A simulation facility is developed using data typical of mid-range saloon cars. While not representing a specific vehicle it incorporates major characteristics which influence fuel and emissions performance, making it suitable for evaluating various engine control strategies.

Simulation studies involving two widely differing schemes for engine control are described in Chapter 6 and Chapter 7. The first is a calibration procedure which does not rely on steady-state engine characterisation, and the second is a simple adaptive ignition timing scheme. The calibration method studied here is similar to an on-line test-bed technique developed at General Motors, which had shown potential in being able to cope with certain system dynamics. The adaptive controller is a 'hill climber' or extremum seeking type, employing a sinusoidal or rectangular perturbation of the controlled variable as a means of determining the relevant plant characteristic; reference is made in this chapter to the simulated performance of such a system for spark advance control during driving schedule tracking.

The final chapter is a critical appraisal of the work and makes recommendations for further research. It has particular relevance to the role of modelling and simulation for the control of automotive systems, and refers to a subsequent important programme of work which has been undertaken by the author; this work involves the development of a dynamic automotive system modelling and simulation environment, and has been strongly influenced by the study that formed the basis of this thesis.

CHAPTER 2

ENGINE CONTROL SYSTEMS OPTIMISATION

2.1 EMISSION CONSTRAINED FUEL ECONOMY.

A particularly important aspect of engine control system optimisation is related to the trade-off between fuel consumption and noxious emissions.

The emission constrained fuel economy problem takes the same basic form worldwide. Where differences exist they are mainly due to the severity of the emissions constraints imposed. The problem statement is simply:

"Determine the control strategy that provides minimum fuel consumption while meeting the emission constraints prescribed for the given driving schedule." (2.1)

Clearly, in view of the trade-off between emissions, fuel economy and hardware costs it is not in a manufacturer's best interests to produce a vehicle which greatly surpasses the required emissions performance. Attempts at obtaining an acceptable balance between emissions and fuel economy may be made in many ways. Redesigned engines (e.g. stratified charge, lean-burn), electronic control of air-fuel ratio (AF) and spark advance (SA), catalytic converters with feedback AF control are all included in the range of possible engine hardware (2.2-2.5).

It is evident therefore that good methods of calibration need to be devised in order to exploit the inherent potential of a given technology. Calibration is a term used in this context to describe the derivation of laws to govern the action of the engine controllers (SA, AF and EGR if fitted) for a particular automotive system. Such controllers ensure the 'correct' value for the engine control variables, dependent on the demanded engine torque and the relevant ambient conditions. The definition of 'correct' or 'optimal' control itself depends upon the emissions and vehicle test legislation; which provides a basis for comparison of vehicle fuel consumption.

Often the weighted effects of more than one driving schedule will be used for vehicle evaluation. If for simplicity we consider economy during a single schedule as the performance metric we can formulate a constrained optimisation problem:

Select the control vector

$$u(t) = \begin{bmatrix} SA(t) \\ AF(t) \\ EGR(t) \end{bmatrix}$$

to minimise the fuel consumption

$$J = \int_{t_0}^{t_f} \dot{F}(t).dt$$

subject to

(1) engine/vehicle dynamics $(\dot{V}, \dot{W}, \dot{T})$

(2) engine description $(\ddot{F}, \ddot{CO}, \ddot{HC}, \ddot{NOx})$

(3) driving schedule Vd

(4) emissions constraints

$$\int_{t_0}^{t_f} f(t) \, . \, dt \leqslant \begin{bmatrix} CO^* \\ HC^* \\ NOx^* \end{bmatrix}$$

(5) driveability constraint $C(t) \leqslant C^*$, $t_0 \leqslant t \leqslant t_f$

where V = vehicle velocity

W = engine speed

T = engine net torque

$\dot{F}$ = fuel flow

$\dot{CO}$ = carbon monoxide emission flow

$\dot{HC}$ = unburnt hydrocarbon emission flow

$\dot{NOx}$ = nitrogen oxides emission flow

f = vector of emission flows $[\dot{CO}\ \dot{HC}\ \dot{NOx}]^T$

and z^* designates the constrained level of variable z

$C(t)$ is some measure related to the 'roughness' of engine operation, and contributes to the subjective 'feel' of the vehicle as experienced by a driver.

Figure 2.1 illustrates a typical engine control system. In a conventional automotive system the spark advance is mechanically controlled by springs and bob weights in the distributor to give ignition advance as a simple function of engine speed. An additional mechanism senses inlet manifold pressure (approximately proportional to engine load) and retards the timing at higher loads. The primary approach to air-fuel ratio control is to sense air flow and meter fuel accordingly. Here a combination of inlet manifold pressure, fixed or variable venturi and air mass flow transducers may be used (2.6).

While mechanical and relatively simple electrical devices have been the only reliable and cost-effective means of control in the past, it is evident that they have their limitations when stricter emissions control measures are imposed and attention is also focussed on fuel economy. Technological advances have made it possible to produce reliable electronic controls that can implement sophisticated algorithms to improve all areas of vehicle performance.

The formulation above gives the bare essentials of the problem. The control vector may also include throttle angle and gear ratio, particularly if a CVT is used in the vehicle.

The problem is also non-linear since engines and actuators are non-linear, as well as the vehicle characteristics; environmental and seasonal conditions, ageing and driving schedule characteristics are time-varying phenomena; and sensor noise, combustion variation, manufacturing tolerances and environmental parameters are stochastic variables (see Table 1.1). This amounts to a formidable problem which is at worst multi-variable, time varying, interacting, nonlinear and stochastic in many respects.

Computers are currently being used extensively to aid engineers in calibrating engines for a wide range of vehicles, using procedures which are based on steady-state information obtained from engines. The use of dynamic systems simulation has opened up new possibilities in the field of fuel economy and emissions control. It enables the automotive systems engineer to evaluate the potential of new approaches to calibration and engine control, even if the hardware or sensors have not as yet been fully developed; it allows early prediction of the effect of parameter changes; and can greatly reduce the frequency of costly post-production changes. The ease with which system changes can be made and experiments repeated precisely, without tying-up expensive test facilities, enhances the value of dynamic system simulation, making it a very attractive tool for the development of new advanced powertrain controls.

2.2 APPROACHES TO THE ENGINE CONTROL PROBLEM.

2.2.1 Introduction.

Two basic approaches to the control of S.I. engines have emerged:

- programmed controls,
- feedback and adaptive controls.

A programmed control schedules the controls in some predetermined manner (open-loop) according to the various sensed variables (speed, load, etc.). The accuracy of this type of control depends on the method of calibration and the number of sensed parameters. The calibration method directly addresses the emission constrained fuel economy problem as stated above.

Two adaptive controllers applicable to automotive engines are the extremum-seeking type (hill climber), and the self-tuning type. An extremum-seeking controller may adjust ignition timing and/or air-fuel ratio to maximise some measure of performance such as engine power. This type of control can easily be employed to minimise fuel consumption but its treatment of emissions may be somewhat incidental. One considerable advantage not shared by the programmed (feedforward) control is the ability of this type of controller to respond to unsensed and stochastic variables. Self-tuning controls can perhaps be viewed as standing on the middle ground between the hill climber type of controller and the conventional feedforward controller: in one form it would have a similar parameterised control law as the latter for open loop SA and AF control, but also have an automatic facility for updating or tuning the parameters as system or ambient conditions drift.

2.2.2 Programmed Control Approach.

a) General.

Engine controls have been predominantly of the programmed type in the past. In the case of ignition timing, speed and load (intake vacuum) have been used in the well-known manner. In the case of mixture controls sophisticated carburettors ordinarily respond to manifold pressure and temperature (and often additional parameters).

Latterly LSI circuits have facilitated more precise scheduling of spark advance, air-fuel ratio and exhaust gas recirculation on the basis of speed, load and coolant temperature, etc. The concept relies on the exact characterisation of the engine (as a function of the state variables) to enable the correct programming of the executor. There are several sensor inputs (but many more relevant parameters) essential for reasonable controller performance.

b) Calibration.

Calibration of the controller is the all important procedure responsible for the performance of the vehicle in terms of fuel economy, emissions, driveability and acceleration for a given engine/vehicle combination. Most of the techniques reported are based on mathematical optimisation using a 'steady-state' description of the automotive engine. In most post-1975 examples the vehicle and drive-train have been simulated (2.7) to obtain a matrix of speed/load points representative of the schedule, or to enable the schedule to be 'driven' on a dynamometer.

Generally 8-15 representative time-weighted speed-load conditions (called 'mapping points') are chosen (2.8-2.14) and fuel flow and emissions data obtained at each point for a range of control settings.

Methods of using this engine map vary:

i) Multiple regression analysis can be performed (2.8, 2.9, 2.12) to obtain an engine model in terms of the control variables. The 'optimum' control can then be selected by direct search (2.12), linear programming (2.8) or non-linear programming (2.9).

ii) Dynamic programming (2.10, 2.14) using a set of engine speed/load map points representing engine usage for a specified driving schedule. The total allowable quantity of each pollutant is seen as a resource to be 'allocated' amongst the (time weighted) mapping points, in such a way as to minimise the fuel consumed; this thus defines the controls at particular speed/load conditions.

iii) Use of the Disadvantage Function (2.13) to link the mapping points after initial allocation approximating that in (ii) above.

iv) Perform the operation on-line with a gradient algorithm (e.g. Powells conjugate direction algorithm) at each map point; linking the points with Lagrange multipliers (2.11).

All of the above methods are useful only for warmed-up engine calibration and they do not account for temperature excursions, actuator dynamics or cold-start phenomena. Tennant et al (2.15, 2.16) obtained a 'temperature history' for the warmed-up engine operating mode by rapidly sequencing the test bed dynamometer system through a set of twenty-three speed/load test points. Six different sequences were used and the data-base was accumulated in no more than twelve hours; significantly faster than the more normal steady-state mapping procedure. The results underlined the sensitivity of HC and NOx to temperature: effects unobtainable from the former entirely 'static' engine mapping data-base.

So far only Dohner (2.17) has reported any attempt to calibrate an engine over the whole of a prescribed driving schedule. He used optimal control theory, dispensing with engine models, and generated the system equations by 'driving' the whole schedule on a dynamometer. The procedure relies on the ability to collect the engine data continuously in order to determine the feedback variables for the next iteration. This approach is quite costly in time and resources but yields a solution which accounts in some measure for catalytic converter, cold-start, actuator and transmission dynamics incorporating also a formal treatment of driveability. This latter is related to the variability of the BMEP for a rough-running engine.

c) Implementational Considerations.

One major limitation of an engine control based on the methods outlined above, is the fact that the optimality of its control function is undoubtedly characterised by specific test conditions. Very often a reasonable control developed as an emission-constrained minimum fuel solution, has to be radically modified in the vehicle, simply because true dynamic vehicle information has not, and in most cases, cannot be incorporated into the calibration procedure.

The open-loop or programmed control has been fitted to the production engine since its inception. The carburettor venturi system provides a fuel/air mixture according to some prescribed schedule; the conventional spark advance mechanism senses engine speed and load and performs in an open loop manner. The advent of computerised data collection and processing, and the development of cheap micro-electronic devices has facilitated vast improvements in the performance of programmed controls. It is now possible to implement far more complex control schedules, in order to closer approach the desired performance. Always, however, a conservative calibration is necessary in order to cope with the unsensed or stochastic variables.

2.2.3 Feedback Control.

Many automotive manufacturers have installed '3-way' catalytic converters on vehicles destined for the Japanese or U.S. market, in order to reduce exhaust emissions to an acceptable level. Hot exhaust gases pass through the device, which has a conversion efficiency similar to Figure 2.2, at operating temperature. It is clear that in order to obtain 'best' converter performance the air-fuel ratio must be maintained very close to stoichiometry. In addition there is a requirement for lead-free fuel in order to avoid rapid degradation of the catalyst.

The open loop AF schedules are unable to perform with sufficient accuracy in this context and closed-loop, or feedback devices have been employed. Using an oxygen sensor in the exhaust stream has permitted systems to be developed based on electronic fuel injection (2.18, 2.19) or feedback carburettors (2.3, 2.5, 2.20). The available sensors (2.21, 2.22, 2.23) have switch-like characteristics around the chemically correct (stoichiometric) mixture ratio; thus restricting their use to limit cycle operation around this air-fuel ratio.

The configuration illustrated in Figure 2.3 is widely used and performs well under normal conditions. The basic form is of a servo which richens or weakens the fuel/air mixture when driven in one or other direction. Owing to the discontinuous response of the sensor to air-fuel ratio and the delay in the system, the servo is driven first in one direction and then in the other under closed loop operation. This results in periodic fluctuations in air-fuel ratio known as limit cycles; the system response is limited generally by the limit-cycle frequency, which depends on the delay due to the dynamics of the engine.

The looked-for improvements in feedback mixture ratio control depend on the development of the necessary sensors. A sensor which has more linear characteristics could be incorporated in a programmed controller to minimise the error between the demanded and actual air-fuel ratios. This would result in faster more accurate controls being implemented on vehicles. Present sensors also must be placed in the exhaust stream where the necessary high temperature can be attained; it is desirable to have a sensor which can measure air-fuel ratio prior to induction of the charge.

2.2.4 Adaptive Control Concepts.

A feedback controller essentially addresses the problem of minimising the effect of state perturbations. Adaptive concepts are oriented towards eliminating the effect of changes in the dynamics of the controlled plant (2.24).

Figure 2.4 (from Reference 2.25) shows an adaptive controller of the self-tuning type which could be used in a feedback air-fuel ratio controller. The controller gains are adjusted to allow optimal performance under a variety of engine operating conditions.

Draper and Li in 1951 (2.26) reported the introduction, to automotive engines, of 'optimilizing' controllers; which belong to a different class of adaptive controls known as 'extremum-seeking' or 'hill-climbers'. Much subsequent work involving this type of controller has been reported by Sweitzer, who has enumerated the merits of adaptive engine control strategies in a number of papers (2.27, 2.28).

The concept of the optimizing controller used in this work is shown in Figure 2.5 and will be examined in detail in Chapter 7. It operates by analysing the effect on engine output power caused by a 'dither' on the control variable set-point, and adjusting this set-point accordingly. The adjustment is made such that it results in a progression towards the maximum output power with respect to the particular control parameter.

Such perturbation controllers have been applied to the control of air-fuel ratio and spark advance of automotive engines. There are difficulties in comparing the performance of these controllers with the more conventional programmed type, as the design aims differ. The latter (programmed controllers) have been designed to directly address the emission constrained fuel economy problem (Section 3.1.2); while adaptive controls have generally an incidental effect on emissions.

The use of simulation techniques early in the development of new or non-standard engine controllers, such as those referred to above, has an added attraction: the risk of incurring serious damage to the engine or equipment can be minimised. The model used as a basis for the simulation must, however, be suitable for the particular design or development work intended.

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NOTATION

AF	air-fuel mass ratio
SA	spark advance
EGR	exhaust gas recirculation
u	control vector
J	cost function
t_0	driving schedule start time
t_f	driving schedule end time
$\dot{F}$	fuel mass flow
Vd	desired vehicle velocity
V	actual vehicle velocity
W	engine speed
$\dot{C}O$	carbon monoxide emission flow
$\dot{H}C$	unburnt hydrocarbon emission flow
$\dot{N}Ox$	nitrogen oxides emission flow
f	vector of emission flows
CO^*	carbon monoxide mass constraint
HC^*	unburnt hydrocarbon mass constraint
NOx^*	nitrogen oxides mass constraint
C	'driveability'
C^*	constraint on 'driveability'

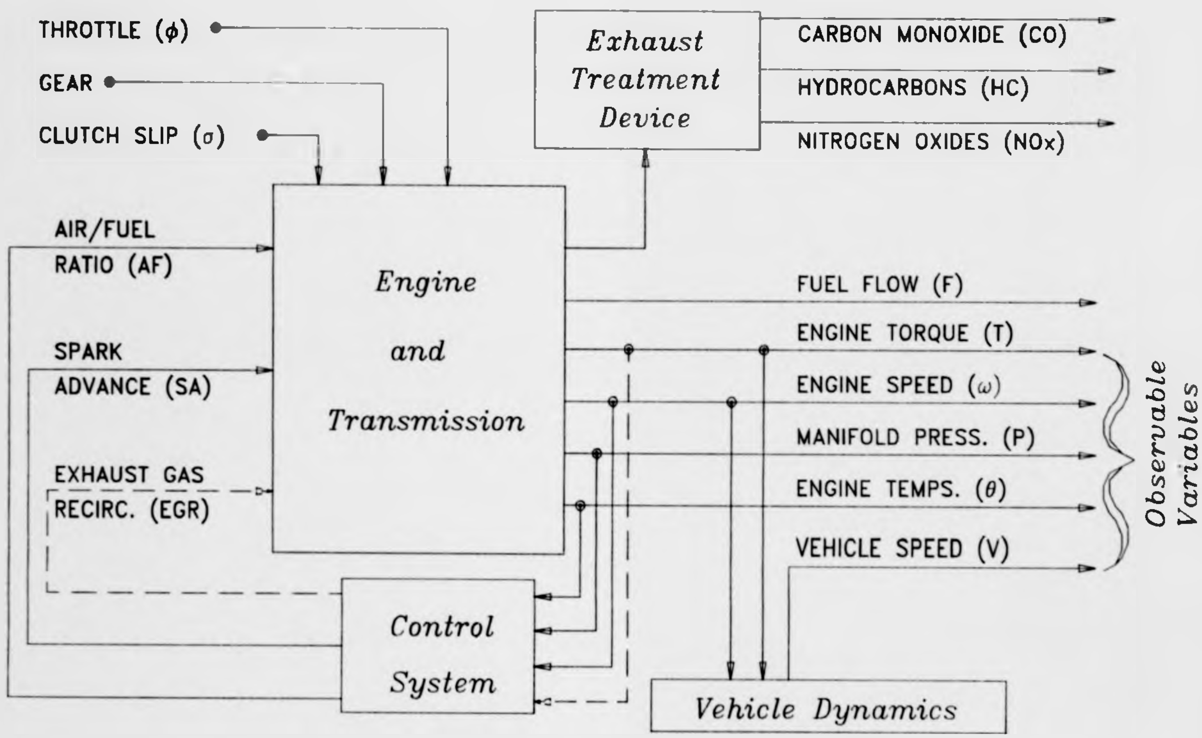

Figure 2.1 Typical Engine Control System

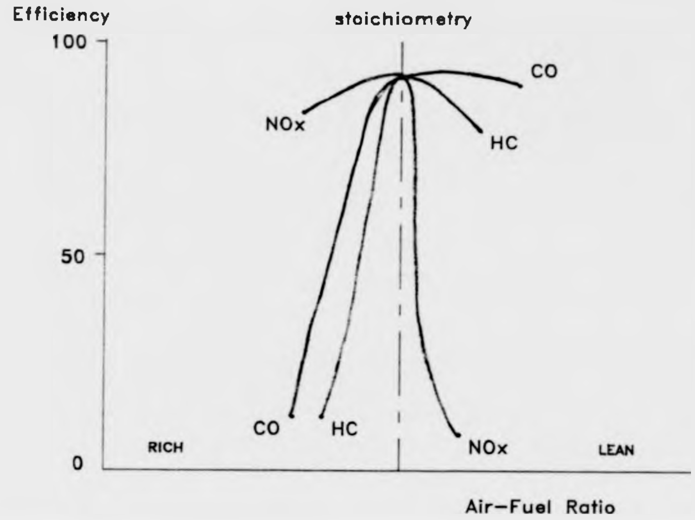

Figure 2.2 Typical 3–Way Converter Efficiency

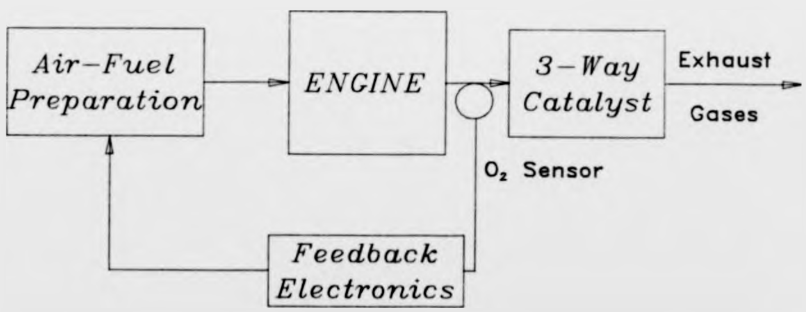

Figure 2.3 Conceptual Representation of a Feedback Engine Air–Fuel Controller

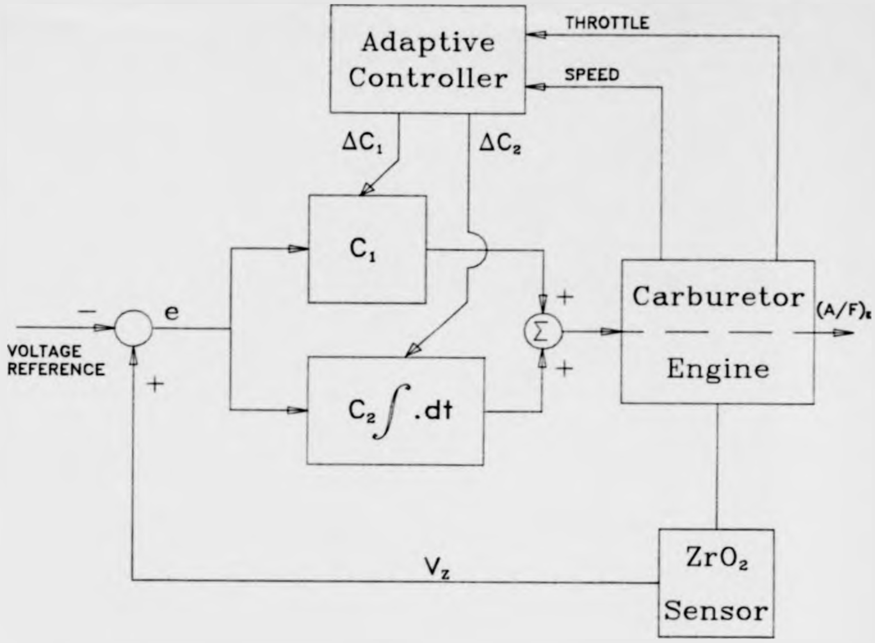

Figure 2.4 Adaptive air–fuel ratio controller compensates for changes in system dynamics (source reference 2.25)

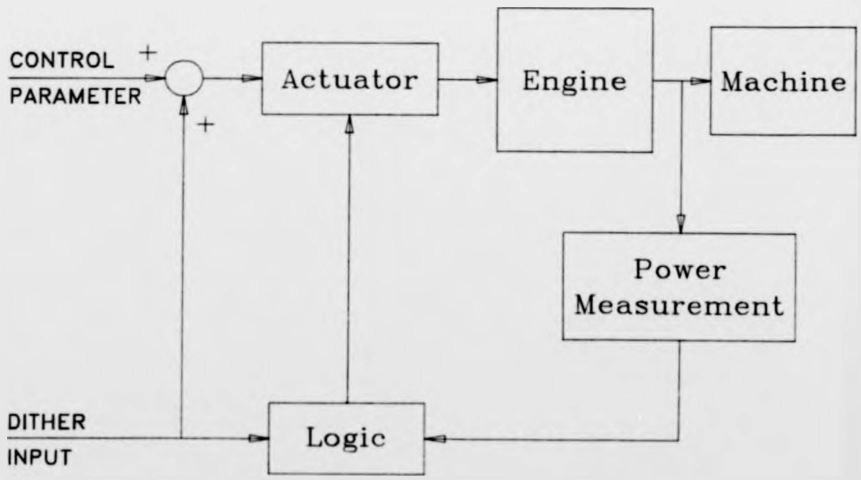

Figure 2.5 The Concept of an Optimizing Controller

CHAPTER 3

BACKGROUND TO AUTOMOTIVE SYSTEM DEVELOPMENT AND MODELLING

This chapter provides background considerations for modelling of automotive systems for fuel economy work: the first section deals with the problems of assessing vehicle performance by means of test procedures; the second section indicates the scope of development, and gives an interpretation of the development potential, of automotive systems; the third section provides a brief description of certain fundamental engine characteristics, which are essential apriori information for modelling for emission constrained fuel economy work.

3.1 A BASIS FOR VEHICLE COMPARISON.

3.1.1 Vehicle Testing.

It is true to say that to many purchasers fuel economy is an important criterion in the selection of a private car. It is an important consideration also to companies who run large fleets of vehicles, saloon cars or utilities. The increasing importance of fuel related performance is highlighted by the contents of the motor manufacturers' promotional material; and for a long time the data was introduced with such nebulous terms as 'overall fuel consumption' or 'touring fuel consumption'.

Manufacturers have always had some means of determining these figures using their own road tests, driving schedules or simulations, but no common procedure was adopted to permit fair comparison of competing products. The problem in comparing published economy figures stems from the fact that they depend greatly on the measurement techniques and physical conditions prevailing during the vehicle test, as well as the nature of the test itself. It was therefore essential that some reliable technique for fuel economy measurement be universally adopted by the automotive manufacturers.

The introduction of vehicle emission controls in the U.S.A. in the later 1960's accentuated this need; and indeed standardised vehicle test schedules had to be developed in order to facilitate the use of emission control measures. Both fuel and emissions performance are intimately related to vehicle duty and driver behaviour; and it is clear that precise and repeatable fuel consumption measurement can only be achieved when all sources of variability are eliminated by the test procedure. This can be a difficult and costly exercise, and may eventually lead to a procedure that gives accurate and repeatable results but is completely divorced from the reality of everyday driving.

The major considerations in devising a vehicle assessment scheme involve the control or elimination of changes due to: road conditions (e.g. urban or rural routes, traffic flow); ambient conditions (e.g. wind, temperature, pressure); and driver behaviour (e.g. the use of cold-start devices, acceleration rates, average speed).

Various methods have been used by industry for fuel economy evaluation as follows (3.1):-

a) Uncontrolled road tests - none of the variables mentioned above are controlled.

b) Controlled road tests - one or more of the variables mentioned above are held constant (excluding cycle tests).

c) Cycle tests on the road (including constant speed tests).

d) Cycle tests on a chassis dynamometer (including constant speed tests).

e) Bench engine tests.

These test methods attempt to reduce some if not all of the sources of variability mentioned above to a greater or lesser degree. The dividing line between driving schedule tests on the road and a highly controlled road test, may be somewhat arbitrary.

The use of chassis dynamometers in vehicle testing gives much greater control over physical conditions than is ever possible with road tests. It means that reliable vehicle comparisons can be made with relatively little expenditure of time and money. The correspondence to actual vehicle usage, however, still depends on the nature of driving schedules or test methods adopted.

The dynamometer provides a load to the driving wheels of a stationary vehicle via large cylindrical rollers. The vehicle can be 'driven' on the equipment and the resistance at the driving wheels is varied according to the acceleration and speed in order to simulate vehicle inertia, aerodynamic and frictional forces. Such parameters are generally simulated on the dynamometer using flywheels and/or electronic controls. The clear advantages of using the chassis dynamometer in fuel and emission tests are limited in practice by the accuracy with which the driving schedule is followed; by the quality of the simulation of the vehicle drag characteristics; and by the degree of correspondence between the engine operating conditions on the dynamometer and those on the road (particularly with regard to temperatures and air flow).

3.1.2 Vehicle Emissions Legislation.

Combustion engines emit complex mixtures of gases, liquids and solids during use (3.2) which have concentrations relating to engine design, operating conditions, manufacturing tolerances, mechanical wear and a number of other factors. Such emissions present a threat, the severity of which depends largely on geographical and meteorological factors. It is as the population of vehicles in society rises that the ecological conditions become as important as those of energy usage. There comes a point where the high concentration of certain gaseous constituents of engine exhaust in the atmosphere becomes hazardous to the health of plants and animals. It was not until the early 1960's that such facts were realised and governments considered the introduction of measures to restrict the production of noxious components.

The situation prevailing in Los Angeles at that time is well known and caused ripples of concern to spread throughout the Western world. It is now thought that the severity of the pollution was due to a unique combination of geographical and climatic conditions, as well as the high density of the private car population (3.3). There is not sufficient medical evidence to define a 'dangerous' level for each pollutant, but it is fair to assume that there exists a level also, that while not presenting a health hazard, will certainly pose some threat to the environment.

Unsure of the full cause of the unpleasant smog over Los Angeles, and uncertain of the health risk involved, the Californian and then the Federal authorities imposed legislation that demanded rather large and arbitrary reductions in concentrations. The Japanese authorities having rather similar problems at a later date did similarly (3.4); but the E.E.C. meanwhile took a conservative stand point imposing milder and more gradual measures, more suitable for our own environmental conditions, and permitting a better appreciation of the effect of such measures (Figure 3.1).

There are three main components of vehicle exhaust that have been subjects of legislation: unburnt hydrocarbons, nitrogen oxides and carbon monoxide. The first two of these pollutants are partially responsible for chemical smog, while carbon monoxide and the oxides of nitrogen are also toxic. Lead and sulphur compounds, though toxic are not dealt with by engine design or control, but at the fuel refining stage. The former is usually added to the fuel in order to improve its octane rating. Particulates in the exhaust are subject to control but the legislature is confined to diesel engines which are not discussed in detail in this work.

3.1.3 Driving Schedules.

A driving schedule is simply a sequence of desired vehicle velocities specified at discrete intervals (often 1.0 seconds). Each schedule is composed of one or more 'cycles' which are separated by periods of zero velocity. A vehicle to be tested will usually be driven on a chassis dynamometer according to a driving schedule velocity profile, while the fuel consumption is monitored and the exhaust gases collected in a special bag (3.5). Gear change times may or may not be specified by the schedule.

The U.S. Environmental Protection Agency (E.P.A.) adopted a set of cycles based on the so-called 'LA-4' test route (Figure 3.2): a route of 12 miles on roads in the centre of the Los Angeles basin. The result was the urban dynamometer driving schedule (U.D.D.S.) of 7.5 miles in length. The first five cycles form the 'cold-start' portion of the test which is immediately followed by cycles six to eighteen, the 'stabilised' portion. After a ten minute 'hot soak' with the engine switched off, the first five cycles are repeated. This forms the 'hot-start' portion of the test, making the total simulated distance travelled equivalent to 11.04 miles (17.77 km). A weighting applied to the emissions produced in each of the three portions gives the overall grams-per-mile of individual pollutants.

In 1975 a further schedule (Figure 3.3) was added to the test procedure and is known as the 'highway' cycle (3.6). This 10.25 mile (16.50 km) cycle is not used in the emissions calculation but is included in the fuel economy measurement to include some information about higher speed operation. These cycles are the basis for dynamometer fuel and emission testing, but are far too complex for track testing (3.1, 3.7).

The European and Japanese test cycles are considerably simpler than the U.S. Federal schedules. The E.E.C. introduced the ECE 15 schedule (Figure 3.4) for evaluating warmed-up vehicle fuel and emissions performance during simulated urban driving. No highway or suburban schedule was introduced but constant speed (90 km/h and 120 km/h) economy figures are obtained instead (3.8). The Japanese legislation is linked to a 10-mode schedule similar in form to the ECE 15.

While it is evident that the simple form of the European and Japanese tests lend themselves equally well to track testing and dynamometer methods, resemblence to normal car usage is not apparent; with their constant accelerations, decelerations and fixed gear-change speeds. Aerodynamic effects on fuel economy are not really measured by these schedules as the maximum vehicle speed is low, and considerable time is spent in an idling condition. Conversely the constant speed tests give no consideration to vehicle inertia, or carburation and temperature dynamics. It is not clear, however, that a simulated rural driving schedule would represent 'normal car usage' any better, though it may appear to be less artificial in form.

In the light of the limitations and doubts about current test procedures much work is continuing in the area of driving patterns, in order to permit more accurate prediction of energy usage of vehicles in the future (3.9). The outcome of such research may be an improved set of driving schedules or a completely different 'yardstick'. Certainly future test procedures must acknowledge the different driving patterns associated with cars of different power-weight ratios. An anomaly of the present scheme is that a high-powered sports saloon must be driven in an identical manner to the smallest city run-about. Clearly this is unrealistic as it forces manufacturers to optimise the automotive system for the test method adopted, rather than for the use to which the vehicle will be put.

Having delineated the short-comings of vehicle tests it is fair to emphasise the fact that the development of methods which provide realistic estimates of fuel consumption is very difficult, particularly for such a large class of vehicles. Many of the stochastic variables influencing true fuel economy (Table 1.1) can be eliminated (as mentioned above) for the purpose of repeatability, but the value of the results will be reduced significantly if insufficient attention is paid to realistic modelling of the major aspects of vehicle dynamics, driver behaviour, road conditions and topography (3.1, 3.10).

3.2 THE AUTOMOTIVE SYSTEM DEVELOPMENT POTENTIAL.

3.2.1 Introduction.

The principal elements of an automotive power system are shown in Figure 3.5. The interactions are simplified for clarity but all the components illustrated are essential parts of the structure of an automobile, with the exception only of the energy storage unit which has a unique place in so-called 'hybrid' vehicles. Each element yields itself to optimisation with regard to overall energy management of the system. Some elements can be examined and developed in isolation, while certain complex interactions necessitate an integrated approach to the development of other elements.

While it is true that the driver's performance in the system can be optimised by better education and training with regard to vehicle operation, it is not a direct consideration of this work. Driver behaviour does indeed have significant effect on fuel economy, but largely this is uncontrollable from the viewpoint of the automotive engineer. Road conditions and environmental parameters are also in this category, so it is the desire of the engineer to design and control the automotive power system in such a way that the effect of such external influences is minimal. The result is hopefully a car which is reliable, cheap to produce and makes efficient use of the energy source over the whole range of operating conditions.

In this discussion of automotive system development potential we are concerned chiefly with fuel economy and factors relating to this. The system (Figure 3.5) fortunately can be divided into three broad areas: the vehicle; the power plant; and the transmission. Several studies have been made to determine the most fruitful areas for future development (3.11, 3.12, 3.13) and these have been reviewed by others (3.14, 3.15). The motor industry over the last decade has repeatedly had to seek short-term solutions to the constraints imposed upon their vehicles; making rapid alterations to system configurations, with bolt-on additions to help rectify the new deficiencies of their existing designs. At the same time fresh inertia has been given to schemes addressing the long term solution to the problems. It is in view of this increased activity and the growth in technology that an overview of automotive system development is given here.

3.2.2. The Vehicle.

The first of the three broad categories to consider is the vehicle. It has been said that the design of each component of the automotive system cannot be done in isolation. The characteristics of the vehicle itself have a primary influence on the performance of the car and hence are of fundamental importance to the choice of power plant and transmission. A vehicle that requires relatively large amounts of energy to achieve a satisfactory performance will be uneconomical on fuel.

The effect of vehicle characteristics can best be seen by the examination of 'road load'. Road load is defined as the tractive force required at the rear wheels to drive the vehicle at a given steady speed. This is composed of components due to the vehicle weight, aerodynamic drag and tyre rolling resistance, and can be approximated by the expression

$$F_r = C_R Mg + \tfrac{1}{2} A_f \rho C_D V^2 + Mg \sin\alpha \qquad (3.1)$$

where M is the vehicle mass; ρ the air density; A_f the vehicle frontal area; V the vehicle velocity; g the acceleration due to gravity; the gradient angle and C_D and C_R the coefficients of aerodynamic drag and tyre rolling resistance respectively.

Considering the contribution of vehicle aerodynamics

$$F_a = \tfrac{1}{2} A_f \rho C_D V^2 \qquad (3.2)$$

we see that the effects greatly increase with speed. As a body passes through air the resistance experienced depends on the mass of air displaced. Since the density of air varies with altitude and temperature the coefficient of drag is usually presented in terms of standard temperature and pressure.

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Overall vehicle aerodynamic drag can be thought of as a combination of five major factors:

1. External vehicle shape and size.
2. Flow interference due to protruding parts.
3. Skin friction or surface resistance.
4. Drag due to vehicle lift forces.
5. Ventilation or internal flow effects.

The external shape or form of the vehicle has the largest contribution to air resistance. The ideal body form from an aerodynamic viewpoint would permit 'streamlined' or laminar flow of air over the vehicle (3.16). Minimising air pressure at the front of the car and suction or turbulence at the rear by using an ideal 'tear-drop' shape, unfortunately could not provide for the requirements of the passenger car. Consideration of the vehicles used for speed record attempts will clearly indicate the need to compromise body shape in order to satisfy requirements for passenger and luggage accomodation, convenience and driver visibility.

Surface drag is due to the frictional resistance resulting from the flow of air tangential to the body. Small imperfections or roughness of the surface have effect due to the viscosity of the boundary layer (the thin layer of air adjacent to the body). Any improvement of body paint smoothness will have no effect on drag at land vehicle speeds (3.17). Most body panel joints, fastenings and bright work are covered by the boundary layer of air and thus affect surface drag which contributes about 9-10% of the total vehicle aerodynamic resistance (3.17, 3.18).

Those vehicle parts which protrude further actually interfere with the flow of air over the body. Such items as external mirrors, windscreen wipers, radio aerials and auxiliary lamps come in this category, along with door pillars, gutters, door handles and external luggage carriers. Underneath the body mechanical parts such as the exhaust system and suspension components have additional and sometimes variable effect.

Any lift force that results from the movement of the vehicle produces a corresponding drag due to the energy being taken from the car body. The lift effect is due to lower mean pressures developed on the upper surface compared to that on the underbody. This situation is brought about on the moving vehicle by a combination of body form (including the contours of the underbody), the ground clearance and the angle of attack of the body. These latter naturally depend on the vehicle load and its distribution as well as the power being transmitted to the driving wheels.

Passenger compartment ventilation has a fairly small effect on vehicle drag, while the major part of internal air flow serves to provide cooling for the radiator and other mechanical parts such as brakes. It is clear that criteria for efficient flow of internal air will be compromised by conflicting considerations of aesthetics, mechanical size limitations and cost.

Empirical studies on typical cars on the U.S. market in the early 1970's (3.17) shows the following distribution of total vehicle aerodynamic drag for normal cruising speeds:

vehicle form	55%
interference	17%
internal flow	12%
surface drag	9%
lift	7%

Here 62% of the total distribution (lift drag and form drag) is a basic function of body shape and size. Hence a major part of the improvement program with regard to aerodynamics has concentrated in body design (3.19, 3.20). The basic form of vehicles has changed markedly over the last decade. Manufacturers have developed the transverse engine concept further, resulting in improved accommodation within the vehicle; and concurrent development has produced roomy vehicles with smaller frontal area (a very important parameter in equation 3.2).

The reduction in drag due to body form has to incorporate detail modifications because of interference caused by the interaction mentioned earlier. Modern cars have much smoother underbodies, and improved flow characteristics due to the introduction of devices such as integral front and rear spoilers, additional detailing which help reduce or control air turbulence.

The problems of engineers responsible for car aerodynamics are unfortunately not 2-dimensional, as the effect of cross-winds for instance have a not inconsiderable effect (3.17, 3.21, 3.22) especially as interaction occurs with the suspension system.

It is very difficult to improve upon the internal flow aspect of the passenger car, so it seems that worthwhile fuel economy benefits are least likely to ensue from work in this area, or indeed from attention to surface finish.

The aerodynamic drag coefficient of passenger cars is usually in the range of 0.35 to 0.55 with most being above 0.40. It can prove exceedingly difficult to reduce the C_D coefficient of a standard car from say 0.45 to 0.40 without major styling alterations.

Considering now the rolling resistance contribution to total road load we have

$$F_R = C_R Mg \tag{3.3}$$

which implies that no variation of rolling resistance occurs with vehicle velocity, but it is linearly proportional to vehicle weight.

The coefficient C_R is roughly 0.01 - 0.02 for radial tyres, and although is assumed constant in many simulation models, it is in fact dependent on operating conditions (3.23-3.30).

Most car tyres are structurally similar being constructed of layers of fabric webbing bonded with a rubber compound. The tyre has strong steel bead wires which hold it to the rim, while additional steel wires often brace the tread to limit lateral deformation under large cornering forces. Three main loss mechanisms are well known in the literature and have been summarised as follows (3.23, 3.25):

1. Hysteresis losses within the rubber and fabric components of the tyre body account for roughly 90-95% of the total losses.

2. Friction losses between the tyre and the road under varying driving conditions amount to about 5-10% of the total.

3. Windage due to tyre movement through the air has the smallest contribution to energy loss, and is about 2%. An aggressive tread pattern will have a slightly higher loss.

Apart from tyre construction and materials many other factors influence the energy loss in everyday use. Some of these are included in Table 3.1 from Reference 3.25. It is difficult to ascertain the exact distribution of the losses between individual factors because of their interaction; a tyre travelling at high speed will have a higher equilibrium temperature and a higher pressure, a different driving force and a different load due to the change in vehicle attitude, than the same tyre operating at slower speed. It is thus that the effect due to say high speed (Figure 3.6) may to some extent be counteracted by the rise in temperature and pressure.

One factor which has a very large effect on the rolling resistance during normal driving is the effect of bends. A curve of small radius driven at even moderate speeds dramatically increases energy loss (Figure 3.7), and even relatively small cornering forces result in a significantly increased fuel cost (3.31). Government fuel economy tests, because of the difficulties involved, are concerned only with vehicles driven on a straight course.

The relative effects of aerodynamic and rolling resistance drag are indicated in Figure 3.8. It can be seen that while aerodynamic drag dominates at high speeds it is almost negligible at low speeds. Rolling resistance however is never negligible but increases only a relatively small amount with speed.

The third part of the road load equation (3.1) is a simple function of grade angle.

$$F_{\alpha} = Mg \sin\alpha + M\dot{V} \qquad (3.4)$$

where α is the angle subtended between the road surface and the horizontal. The second term included here is the only one excluded from the steady state equation (3.1). Its inclusion is necessary to account for the force needed to accelerate a vehicle in mixed mode driving. The energy required to accelerate a vehicle is often referred to as 'inertia loss'. This description is inaccurate however as all the energy is still available to overcome aerodynamic and rolling friction losses: it has merely been converted into kinetic energy in the case of the acceleration term, and into potential energy associated with the gradient term.

It would thus appear that the mass of the vehicle would have little effect on economy unless the brakes were applied. This is true to a certain extent since the energy is only lost by conversion into heat when the brakes are used. For a heavier vehicle more energy will be lost in this way however since the driver will still expect it to decelerate at the same rate as other vehicles of its class.

Another major reason for the strong influence of weight in fuel economy is the fact that the size and characteristics of the power plant and drive-train are directly affected. Owners will expect a reasonable performance from their car regardless of the topography; and the power needed to climb hills at normal cruising speeds is much greater than that required to overcome aerodynamic drag or rolling resistance. A considerable improvement in fuel economy is achieved for the same performance level with a lighter vehicle with a matched transmission and engine (Figure 3.9, Reference 3.32).

As is often the case an improvement in one area is detrimental to another; a balance must be struck between vehicle weight, ride quality noise, and vibration. Many of these characteristics in the past have been almost synonymous with vehicle weight. But now the development of suspension technology and new materials and manufacturing technology has enabled lighter, quieter vehicles with the necessary strength, safety, and ride quality to be produced.

TABLE 3.1

Some factors which influence tyre power consumption.

Tyre deflection	Tread pattern
Load	Tread thickness
Inflation pressure	Slip angle
Ambient temperature	Vehicle speed
Suspension rates	Driving torque
Wheel alignment	Braking torque
Tyre materials	Road surface
Tyre design	Topography
Polymers	Road temperature
Construction type	Irregularities
Cord	Surface sharpness
Cord angles	Wheel rim width
Body thickness	Tyre diameter

3.2.3 Automotive Engines.

Hughes (3.14) amongst others describes the evolutionary development of automotive power systems as we know them today; from early beginnings in the steam era. Until about the year 1900 the vast majority of road vehicles used either steam driven engines or battery-electric power units.

The 4-stroke internal combustion engine invented by Otto in 1876 did not gain superiority until after the First World War. And it is probably significant that the first mass-produced car, the model 'T' Ford, was driven by such an engine. Though Diesel invented his compression-ignition engine only some sixteen years later than the Otto cycle engine was introduced, it never really matched the success of the latter in the private car industry.

The forty to fifty years after the first mass-produced engine saw a gradual improvement of the power-weight ratio and an improvement in the specific fuel consumption. Growth in the petro-chemical industry, resulted in better quality fuels and the development of knock-reducing additives, permitting further power and economy increases. By the mid to late 1960's the additional problem of pollution control had come to the fore (Section 2.1.2), to be considered alongside the fuel economy factor.

The coincidence of energy and environmental conservation gave us the problem of reducing the energy consumption and pollutant formation in our engines. At the same time, as technology has advanced in many related spheres, it may become economical to build new power systems using different operating cycles; or to adopt an approach using hybrid techniques.

Among the alternative engines considered over the last two decades are the Rankine cycle (steam engine), the Stirling cycle, the Brayton cycle (gas turbine) and the Diesel cycle engines. Of these the steam engine can no longer be considered a contender on the basis of fuel consumption. The Stirling engine also has failed to come up to emissions or fuel consumption expectations, and is unlikely to be in volume production until the 1990's if it were possible to overcome the many difficulties it faces (3.33).

The gas-turbine engine shows some promise and is worthy of the continued effort in its development. The main problems associated with its use in personal transport are due chiefly to the high temperatures and need for fast dynamic response. Computer predictions indicate that fuel economy increases almost in proportion to the elevation of turbine inlet temperature (3.34). Ford predicts that their AGT-101 research vehicle is intended to provide an improvement of 59% over the petrol engine equivalent over the Combined Federal Driving Schedule.

The success of this engine depends greatly on the development of suitable ceramic materials able to withstand the critical stresses of a turbine blade at temperatures in excess of 1300°C. The use of typical aircraft engine metals had to be excluded on the grounds of their exceedingly high cost. The response aspect of the engine has been addressed by at least one manufacturer (Chrysler) by using a 'response-assist flywheel' and a continuously-variable-transmission of the Van Doorne type (3.34); this basically amounts to a hybrid vehicle. The U.S. government despite these difficulties anticipates volume production by the year 1990.

It would seem that for the foreseeable future the private car will be driven almost without exception by reciprocating engines such as we have used for years. One advantage of the alternative engines mentioned is their tolerance of poorer quality fuels. The petrol or diesel engines currently available are not very fuel tolerant, but for many years to come there can be no realistic alternative to hydrocarbon fuels.

The diesel cycle engine is currently the only real contender with the spark ignited petrol engine. A fuel economy advantage of around 25% is usual for the light-duty diesel engine, partly due to: the higher energy content of the fuel; the increased thermal efficiency due to the increased compression ratio; the leaner air-fuel ratios of the diesel, and the reduction in pumping losses at part load owing to the absence of intake throttling. A number of factors have limited its use in practice and necessitate further development of the engine. Apart from the higher cost, lower power/weight ratio and higher noise levels the diesel engine cannot currently meet the proposed emissions limitations both on particulates and gaseous constituents of the exhaust.

As far as the diesel manufacturers are concerned the legislation presents a severe challenge to the current technology. Many of the measures adopted in gasoline engines such as 3-way catalysts and exhaust gas recirculation cannot work adequately on diesel engines (3.34), while a suitable particulate trap has not yet been devised.

The development potential of the Otto cycle engine is of great importance when considering the viability of alternative engines for future use. The question to ask is: "Can the petrol engine maintain its lead in the passenger car market by meeting all the proposed emissions limits, yet, retain its cost-performance advantage?"

The petrol engine does have a combination of advantages not found in other engines. These have been summarised by Forster (3.35):

> "... high brake mean effective pressure and high maximum speed; therefore high torque and high power-to-swept volume ratio; low weight and low bulk volume for a given power demand; smooth and low-noise operation; low cost per unit of power output; and ability to operate with different fuels such as gasoline, LPG, alcohol (methanol and ethanol); NG, LNG, hydrogen."

Developments of the conventional engine are directed towards four broad areas:

1. Improved air-fuel mixture preparation and induction (3.36-3.41).

2. Better control of combustion to ensure complete burning of the mixture (3.42-3.47).

3. Improved engine controls to ensure correct air-fuel ratio, ignition advance and exhaust gas recirculation for all possible combinations of engine speed and load.

4. Attention to exhaust after-treatment to reduce emissions.

The revolution in micro-electronics technology has played a vital part in the development of improved automobile engines (3.48-3.52). Sophisticated programmable electronic circuits are now implementing complex control functions to give improved fuel and economy performance. Such controls are essential to the most recent work on transmissions and hybrid vehicles, designed to operate the engine always at the most efficient combinations of speed and load (3.53-3.54). These tasks are difficult and the subject of much attention at present, as the conventional engine-transmission system has very poor part-load fuel economy (3.55).

The improvement of our present systems to achieve better economy and emissions under normal operating conditions has seen some encouraging success (3.46, 3.56). Modifications improving cylinder charging at low loads by good cylinder disabling schemes have given better economy (3.46, 3.57-3.58) while other means of reducing throttling losses, such as intake valve power control, have sometimes met with disappointing results: in this case the losses in mechanisms for implementing the fuel saving control are chiefly responsible (3.59).

While each advance in engine design contributes to the overall improvement in fuel economy, it is almost always the case that an improvement in one aspect of the engine relies on technological developments in other areas. This is particularly true where performance is limited by materials or sensor technology. Potentially excellent controllers have been devised but simply cannot be implemented at present due to our inability to measure the pertinent variables (3.60).

3.2.4 Transmissions.

Automatic transmission systems favoured in the U.S.A. are not very efficient generally, due to the losses in the torque converter. Efficiency in the cruising mode can be improved by using 'lock-up' converters, but urban fuel economy remains low. The idea of using more than the usual three or four gears in an automatic gearbox should improve economy noticeably.

The European transmission systems are usually of the manual type with four or five gears. These have high efficiencies, but as with any system the ratios must be matched to the vehicle and power-plant characteristics.

The draw-back of manual and automatic transmissions currently used is that the engine rarely operates at its most efficient (3.55) speeds and loads. This is partly due to the limited number of ratios and also to driver demands of reasonable torque in each gear. With an electronically controlled continuously variable transmission (CVT) the engine could spend most of its time in economical modes of operation. There are difficulties however: if the acceleration capabilities are not to be impaired then a large range of ratios is needed; this means extra cost and lower transmission efficiency. Further, if the engine is always at its most efficient operating point under steady-state vehicle conditions, then any demand for acceleration will necessitate a change in engine speed, to provide the required torque; which is undesirable from a driver's point of view as this introduces a lag in response. Clearly some compromise would have to be sought in the controller design.

While certain CVT's may give only small advantage in a particular vehicle due to their cost and reduced efficiency, they are essential to many hybrid vehicles where energy is stored in a flywheel.

It is evident that advances in many areas of technology and control theory will have a bearing on the performance of future vehicles. Several attempts have been made to predict the potential energy savings of our automotive systems (3.13, 3.32, 3.61, 3.1 p.42). Up to 50% improvement is anticipated over mid to late 1970's levels, with over half of this gain from engine and transmission considerations.

3.2.5 Sensor Technology.

a) Sensor Design Criteria.

The application of modern control theory and optimisation techniques to characterisation and to the constrained fuel economy problem is baulked to a large extent by the inability to measure many of the variables. Even the simplest control systems could be improved if there was a cheap and reliable air-fuel ratio sensor for instance; while the list of 'desired' sensors for research and development purposes seems unbounded. Table 3.2 gives a selection of the most useful parameters to be sensed.

Each sensor of necessity has its own set of design criteria dependent on its use. However every engine control system or subsystem must meet certain basic requirements for automotive use. Basically this means that the component or sensor must be able to endure at least 50,000 miles of normal use for it to be generally acceptable.

This is no small challenge as it is recognised that the environmental conditions under the bonnet of a car are considerably more severe than those of, say, an aircraft or spacecraft, both mechanically and electrically. The most outstanding factor is probably due to temperature extremes, and particularly the rapid temperature cycling (3.62) which gives rise to thermal fatigue. High humidity in conjunction with the temperature effects has also led to many early failures in the past due to corrosion and other humidity related failure mechanisms. A recognition of the mechanical and electrical stresses endured by automotive electronics, is given by the test requirements of a typical electronic ignition module (Figure 3.10).

Despite the stringent reliability and fail-safe criteria, sensors and electronic systems must not be costly. This leaves no room for complex and expensive redundancy and protection techniques as are used in aerospace and military fields.

Major problems that need to be overcome in measuring many of the parameters of Table 3.2 relate to the dynamic response and stability requirements. Some variables are difficult to measure accurately and rapidly even in the laboratory: the exhaust emission constituents are in this category (3.2, 3.5, 3.47, 3.63). There are no sensors available that will analyse exhaust gases to determine pollutant partial pressures effectively in a car; neither are there any that have the dynamic response and low-drift characteristics desirable for engine characterisation.

While it might be argued that, although desirable, it is not essential to have on-board measurement of exhaust emissions, it certainly is essential to sense variables associated with fuel management. Any engine control system must also be able to respond to variations in ambient conditions as well as the demands of the driving schedule. Failure to do so will result in degraded driveability, fuel economy and emissions performance.

It must now be clear that in order to achieve the goal of a control that schedules air-fuel ratio and spark advance in an optimal manner, a proliferation of engine sensors will need to be incorporated. Here 'optimal' refers to the ability of the control to give 'best' fuel economy and acceptable driveability and emissions performance. The capability of 'adaptive' controls to respond to unsensed parameters is one point in their favour, and was discussed in Section 2.2.3.

b) The Sensors.

Crankshaft position is a necessary parameter for ignition control, and sensor technology is now well developed for its measurement. Various techniques utilising magnetic phenomena or optical sensing are used, though the latter can suffer from contamination problems (3.64). The sensors easily double for engine speed measurement also and can be used to derive some estimate of engine roughness or driveability (3.65).

Position measurement transducers such as are needed for throttle angle sensing are predominantly potentiometric. Commonly the ceramic or plastic element types are used for long and reliable service.

Temperature sensing is another area where relatively mature components can be used. Wire-wound and thermistor types have long been used and new, faster sensors are being developed for new control applications on automotive engines (3.66, 3.67).

Fuel management systems invariably need sensors to regulate the fuel-air mixture. There are basically three types of control (3.68):

1. Air sensing/fuel metering.
2. Fuel sensing/air metering.
3. Air and fuel programming.

The first is the primary approach of the industry today. The driver adjusts an air throttle: the air is measured and fuel is metered into the air stream accordingly.

The second approach not commonly used in S.I. engines reverses the roles: air is metered as a function of the fuel flow set by the driver.

The third method uses a torque or speed command signal from the operator and calculates both fuel and air quantities to be delivered to the engine. This latter is analogous to the 'fly by wire' concept used in aircraft.

Fuel and air flow measurement presents considerable problems. Fuel flow can be measured relatively accurately if timed discharge fuel injectors are used with a regulated supply pressure. With a carburettor, even if fuel flow can be controlled electronically, it is difficult to meter fuel accurately. Present fuel flow sensors may be incorporated to make accurate steady-state flow measurement/control possible. Their dynamic behaviour is not currently very good and could never assess accurately the fuel entering the combustion chambers owing to the dynamics of the carburettor and inlet manifold.

Air flow sensors would be of great value if sufficiently developed for engine controls. Various concepts have been tried, but most are still in the prototype stage. The chief problems of making a sensor with sufficiently rapid response are: the large dynamic range (typically 30:1); the accuracy required (2%); and the problems of dust, dirt, humidity and mechanical stress. Only one sensor (the moving vane type) has been used for some years in passenger cars (3.37, 3.38).

Because of the lack of mass air flow sensors the usual systems employ speed/density concepts. Manifold pressure and temperature sensors are used to calculate air flow based on the known engine volumetric efficiency at particular engine speeds. Pressure sensors for this purpose are in production but effort is being made to reduce the unit cost. Such sensors also give a fair indication of the engine load, in the absence of torque transducers. Mechanical manifold vacuum operated devices have been used for years in this mode for control of spark advance in relation to load.

A true air-fuel ratio sensor with the necessary dynamic performance and accuracy would solve a number of problems in electronic fuel management. Unfortunately no such device exists, but we do have zirconium and titanium dioxide oxygen sensors in production. These devices measure the oxygen partial pressure in hot exhaust gases and have switch-like characteristics around stoichiometry (chemically correct). High temperatures are required for operation, restricting their location to the exhaust manifold (3.69, 3.70). Engine systems incorporating 3-way catalysts for exhaust after-treatment have a fundamental requirement of a stoichiometric mixture; the use of ZrO_2 and TiO_2 sensors has allowed close control of mixture ratio by modulating the duty-cycle of a controller (3.37, 3.71-3.74). Efforts are now being directed towards developing sensors which have a range extending away from stoichiometry to enable programmable closed loop control at the more economical leaner mixtures.

To maintain the accuracy of calibration a fuel management system must be able to respond to ambient conditions. Atmospheric pressure, temperature and humidity have a marked effect on cylinder charging and hence fuel economy, emission performance and driveability. Temperature of the inlet air is often controlled thermostatically using air inducted near the hot exhaust manifold. No production sensors are yet available for humidity measurement.

It is likely that production torque transducers will soon be available, and these will be a great improvement over manifold pressure transducers for load sensing. The measurement of EGR rate is generally calculated on the basis of ambient absolute pressure and the position of the EGR valve pintle. Such a system, though adequate for stoichiometric mixtures, is unsuitable for lean mixtures where excess oxygen is present in the recirculated gases.

Several commercial sensors have been developed to detect the onset of detonation within the cylinders, whereupon ignition timing is retarded by the controller. These devices are particularly useful for turbo-charged engines, and also permit higher compression ratios in conventional engines. More accurate cylinder pressure sensors have been used to implement adaptive controllers and measure cycle to cycle pressure variations. This latter use can contribute to a 'driveability' evaluation, as combustion stability is directly responsible for torque variations causing roughness, hesitation or even stalling.

TABLE 3.2

Engine Variables Useful in Engine Optimisation and Control.

Engine speed and crankshaft angle	
Engine output torque	
Throttle angle	
Air flow	
Fuel flow	
Air-fuel ratio	
Emissions:	carbon monoxide unburnt hydrocarbons oxides of nitrogen
Ambient conditions:	air temperature air pressure humidity
Engine temperatures:	coolant inlet manifold exhaust manifold catalyst cylinder surface
Engine pressures:	inlet manifold exhaust manifold cylinder
Knock (detonation)	
Exhaust gas recirculation (%)	
Driveability :	surge hesitation etc.

3.3 FUNDAMENTAL ENGINE CHARACTERISTICS.

The aim of this section is to briefly outline engine characteristics regarding factors which influence emission production, fuel consumption and driveability. It is assumed that we have an S.I. engine of mature design.

The three gaseous components of engine exhaust that are subject to government legislations are the combustion-generated emissions: carbon monoxide, nitrogen oxides and unburned or partially burned hydrocarbons.

Carbon monoxide is always present in engine exhaust gases due to dissociation, but chiefly it occurs when there is a deficiency of oxygen in the fuel air mixture (i.e. rich mixtures).

Nitrogen oxides (NO, NO_2, N_2O_2, etc. commonly referred to as NOx) are formed at high temperatures by dissociation of molecular oxygen and nitrogen. The combustion products are cooled rapidly during the expansion stroke yielding quantities of NOx far above that predicted by the equilibrium equation, because of the slow rate of reaction at lower temperatures. Factors which tend to give higher combustion temperatures give higher levels of NOx, while the variation of NOx with air-fuel ratio gives a typical bell-shaped curve (Figure 3.11).

The presence of hydrocarbons amongst the combustion products indicates that combustion has been incomplete. A principal cause is the so-called quench layer adjacent to the relatively cool cylinder walls. The thickness of this layer is a function of air-fuel ratio, pressure and cylinder surface temperature (3.2, 3.63). Much of this layer is burnt during the expansion stroke and part of that remaining is combusted in the hot exhaust manifold; depending on the excess of oxygen present, the temperature of the manifold and the residence time of the gases in the manifold. High combustion temperatures tend to reduce hydrocarbon emissions, and good mixing of the charge ensures more complete combustion. The increase in hydrocarbons at very lean mixtures (Figure 2.10) results from combustion instability, and 'lean-burn' engines are designed to give better flame stability in this operating region.

Given a vehicle that has no special emissions controls the effect of certain design and operating variables on emissions can be summarised (Table 3.3). Chapter 5 of Reference 3.63 contains a good description of these effects. From the control viewpoint it can be said generally that spark retard from MBT spark timing reduces HC and NOx, worsens fuel consumption but has little or no effect on CO. The effect of air-fuel ratio has been described above with reference to Figure 3.11. A third control variable, exhaust gas recirculation, though not presently used in Europe, has found use in the U.S.A. and Japan. It is a primary NOx reduction agent which feeds some of the exhaust gases back into the inlet, thus reducing peak combustion temperatures by charge dilution. The major problem with EGR is the reduction in driveability which often necessitates a richer mixture.

TABLE 3.3

Effect of Design and Operating Variables on Exhaust Emissions and Engine Air Flow. (Source: Reference 3.63)

Variable Increased	HC Conc.	CO Conc.	NO Conc.	Intake Mass Flow Constant Load
Air-fuel ratio	See	Figure	3.8	+
Load	0	0	+	+
Speed	-	0	+-	+
Spark retard	-	0	-	+
Exhaust back pressure	-	0	-	+
Valve overlap	-	0	-	+
Intake man. pressure	0	0	+	+
Combustion chamber deposits	+	0	+	0
Surface/volume ratio	+	0	0	0
Combustion chamber area	+	0	0	0
Stroke/bore ratio	-	0	0	+
Displ. per cylinder	-	0	0	0
Compression ratio	+	0	+	-
Air injection	-	-	0+	+
Fuel injection	-	-	+	0
Coolant temperature	0-	0-	+	0

0 negligible change
\+ positive change
\- negative change

Another characteristic inherent in an automotive system is 'driveability'. A vehicle would be said to have poor driveability if it exhibited a marked unevenness of power delivery due to interaction between components of the system, or if it responded irregularly to external conditions or normal driver inputs. Driveability can be improved by better engine, controller and transmission design or matching; usually a trade-off exists between driveability and fuel and emissions performance. It is difficult to arrive at a rigorous definition of driveability as it is to a large degree subjective.

Efforts directed towards controlling emissions have resulted in a number of design alterations or additions to our engines. However due to their complex interaction it is difficult to attribute fuel economy gains or losses to individual control measures. Despite the difficulty certain basic observations have been made:-

1. For fixed performance and technology, fuel economy deteriorates greatly with tighter emission controls, particularly NOx.

2. For fixed performance and emission levels the fuel economy greatly improves with increased control technological complexity (hence cost).

3. A balance must be sought between performance and fuel economy for given technology and emissions restrictions.

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NOTATION

V	actual vehicle velocity
F_r	vehicle road load force
C_R	rolling restance coefficient
M	vehicle mass
g	acceleration due to gravity
A_f	vehicle frontal area
C_D	vehicle aerodynamic coefficient
ρ	air density
α	grade angle
F_a	vehicle aerodynamic force
F_R	vehicle rolling resistance
F_α	grade force

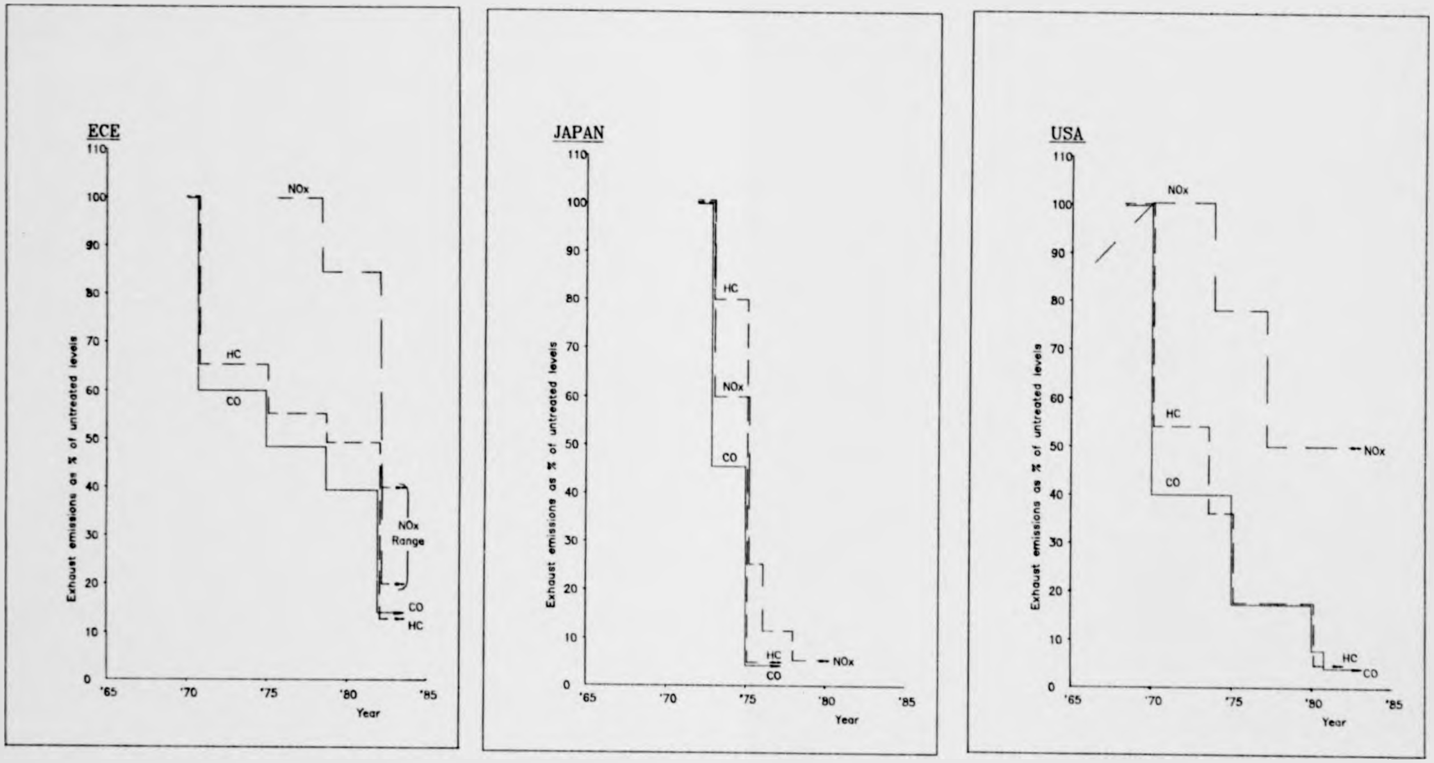

Figure 3.1 Exhaust emission reduction as required by legislation

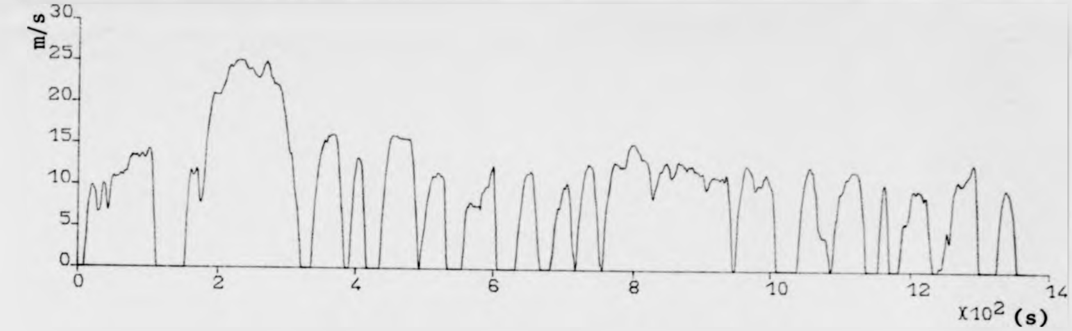

Figure 3.2 EPA Urban driving schedule

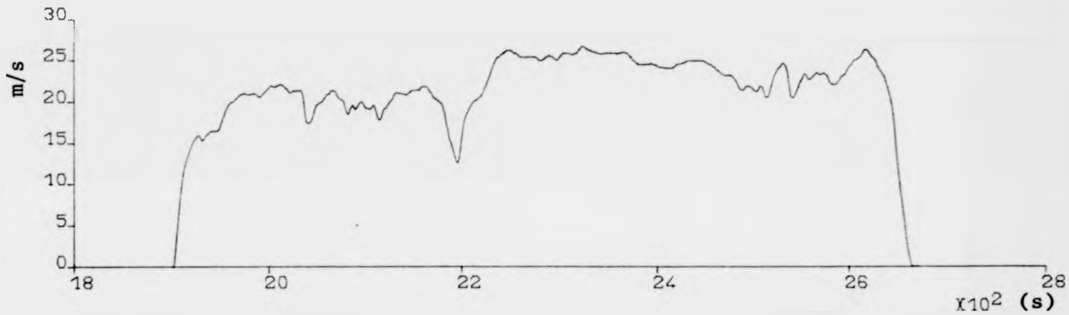

Figure 3.3 EPA Highway driving schedule

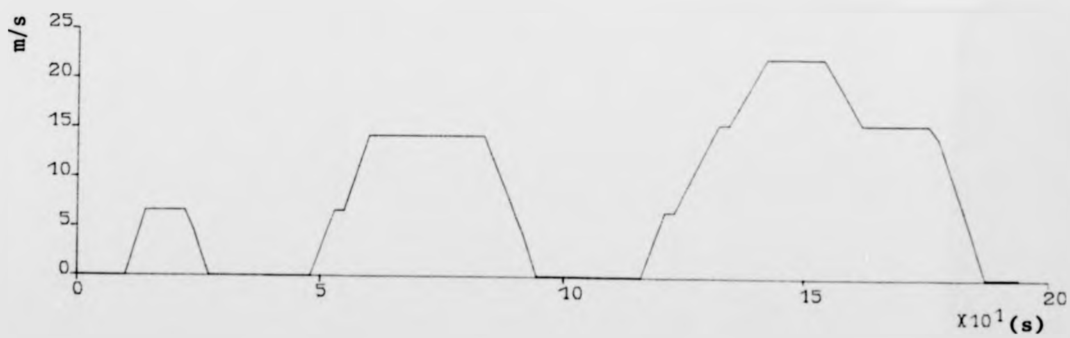

Figure 3.4 ECE−15 Simulated urban driving

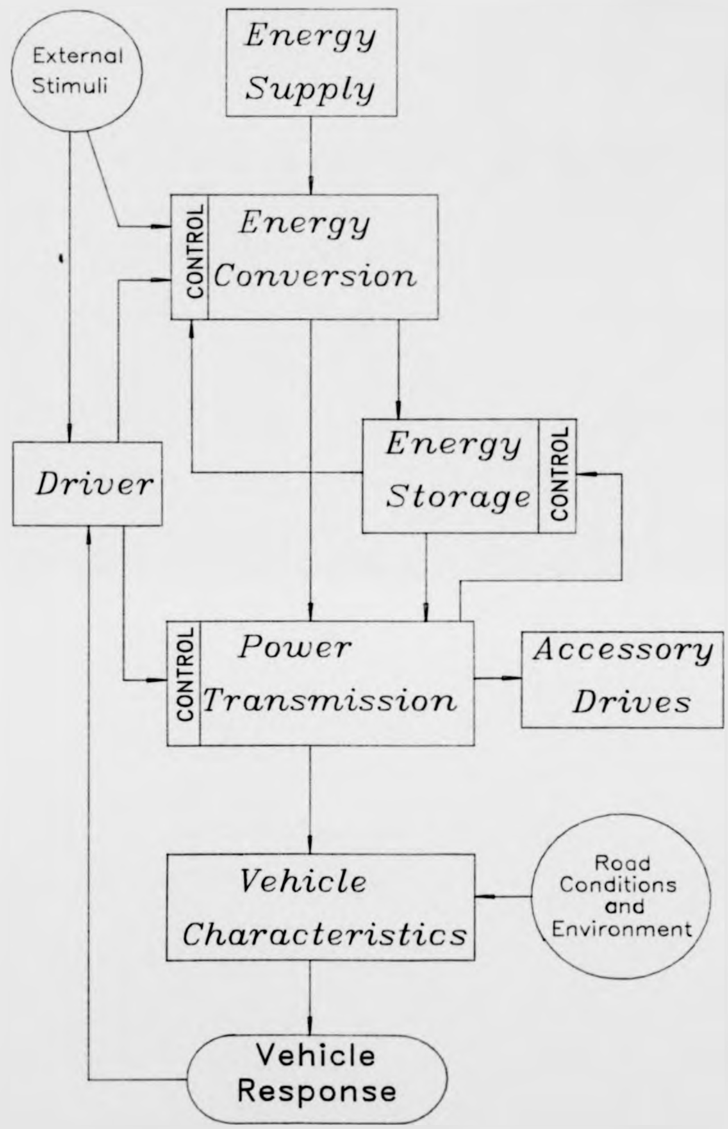

Figure 3.5 Principle Elements of an Automotive Power System

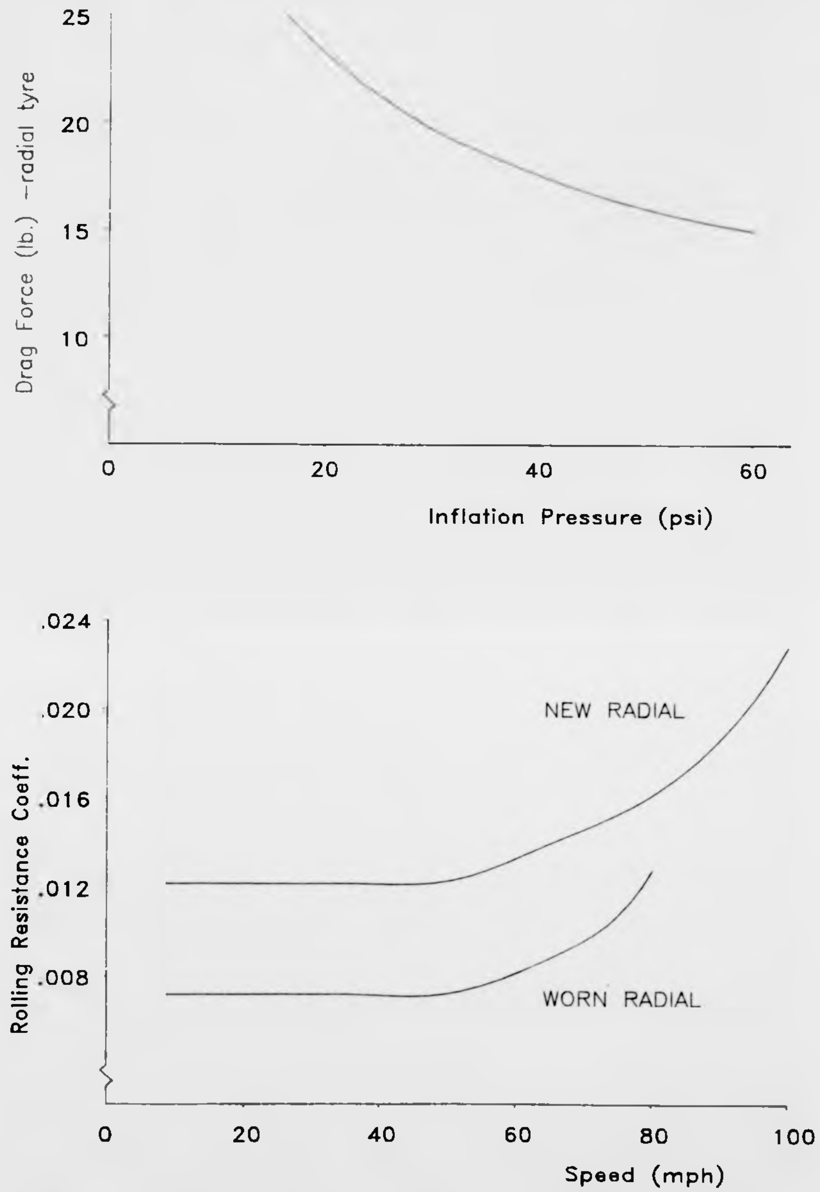

Figure 3.6 Speed and Inflation Effects on Tyre Performance (Source: ref 3.23)

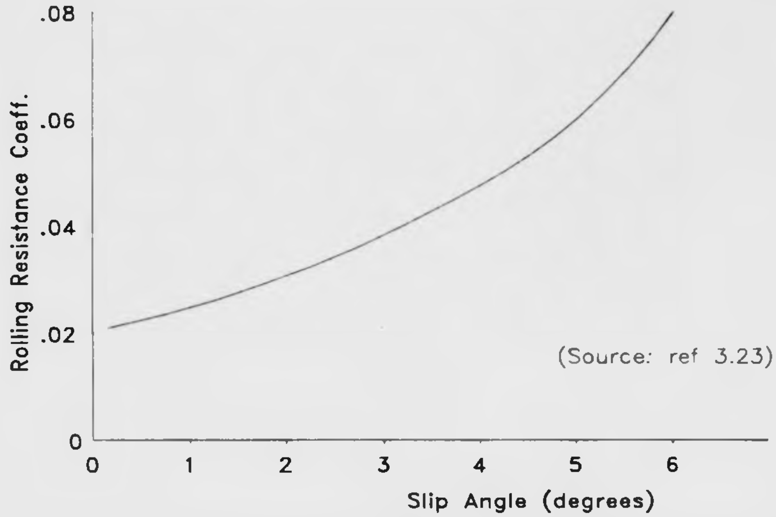

Fig. 3.7 Slip Angle Effects on Rolling Resistance

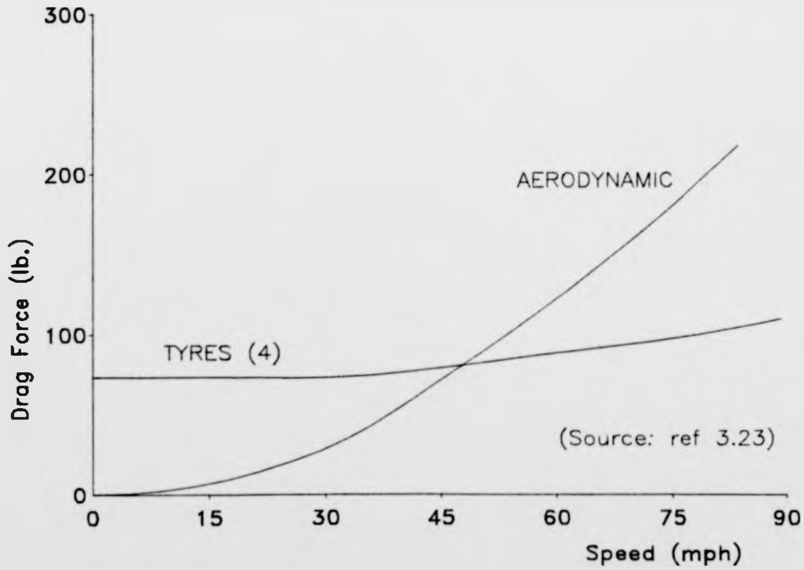

Fig. 3.8 Comparison of Typical Aerodynamic and Rolling Resistance Forces

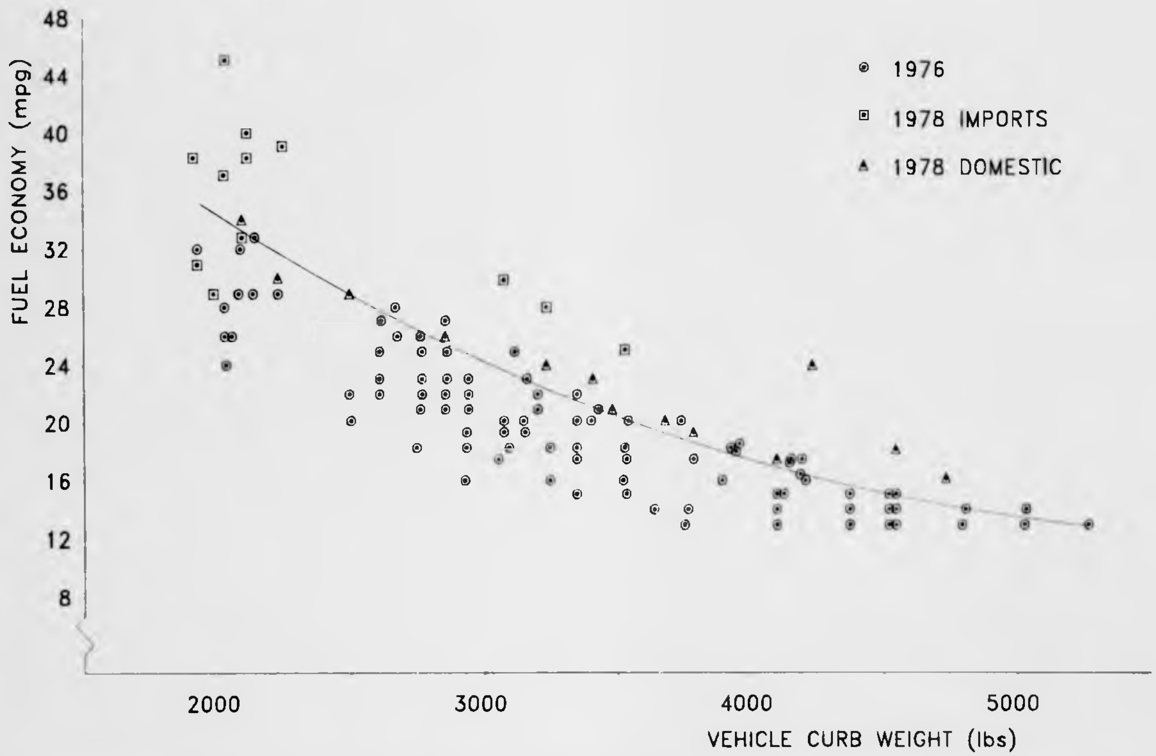

Figure 3.9 U.S. car fleet fuel economy v vehicle weight (Source ref 3.32)

OPERATING TEMPERATURE	*100° to −29°C continuous* *125° to −29°C intermittent*
TEMPERATURE SOAK	*125° and −40°C*
VIBRATION	*30g in 3 planes for 10 hours at frequencies from 50Hz to 300Hz*
REVERSED BATTERY POLARITY	*5 mins survival with subsequent operation unaffected.*
OVER VOLTAGE	*24 volts for 10 mins at 100°C*
TRANSIENTS	*(a) Load dump :−* *Battery disconnected 5 times; or application of an equivalent circuit producing a 120V transient decaying to 1V in 200mS.* *(b) Peak primary voltage :−* *300 volts positive* *(c) Field decay :−* *75 volts negative*
FLASHOVER	*Module to survive all terminals being connected to the high voltage secondary output (20/30 kV)*
DURABILITY	*300 hours at temperature extremes of −29°C and 100°C*
ENVIRONMENTAL	*Module to survive heating to 100°C and immersion into a water/ethylene glycol solution at 25°C 25 times. Also immersion in 5% detergent solution for 96 hrs. at 38°C. Also 96 hrs. salt spray.*

Figure 3.10 Environmental and survival test requirements for an electronic ignition module

(Source ref. 3.25)

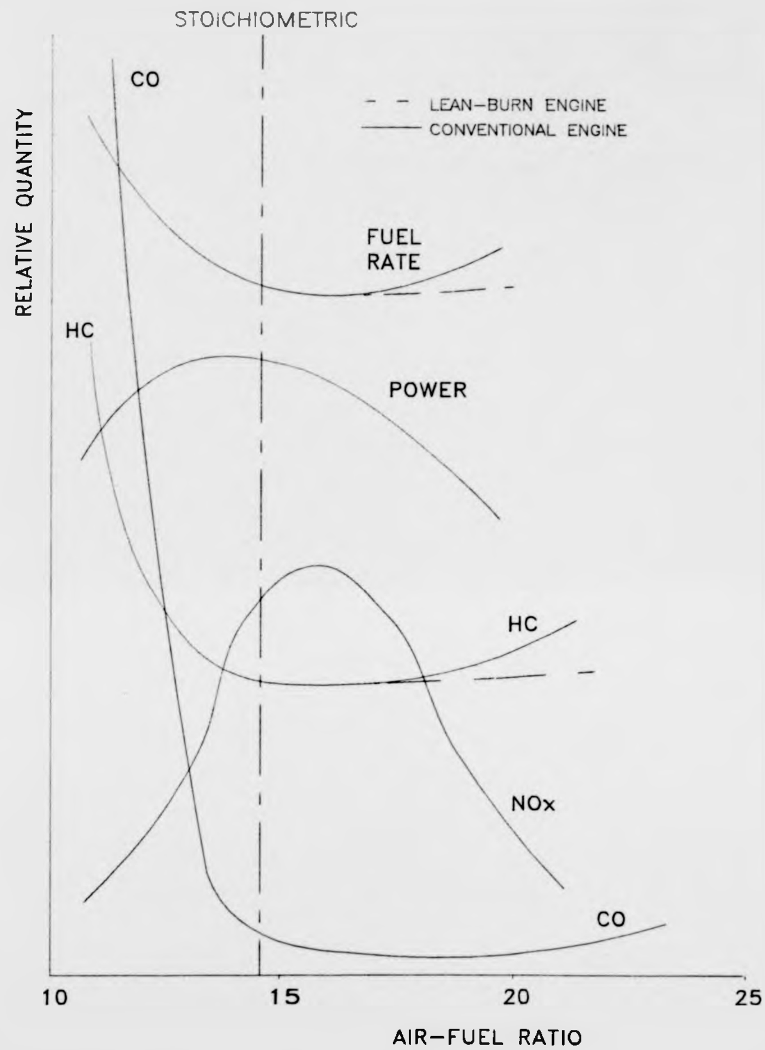

Figure 3.11 Typical emissions and performances relative to air-fuel ratio

CHAPTER 4

VEHICLE POWERTRAIN SYSTEMS SIMULATION

4.1 INTRODUCTION.

In order to develop an adequate representation of any particular system the modeller must first obtain specification of the scope of the simulation tasks to be performed. Without this knowledge he cannot expect to model each element with the appropriate level of detail, in order to efficiently describe the system. The modelling itself is usually a combination of *a priori* knowledge, the application of physical laws, and the use of data obtained by experiment.

The simulation content of this research program required the development of an engine/vehicle model that could be used as a tool in the investigation of engine control problems, such as described in section 2.1. The desire was to use it to study aspects of the performance of optimal and adaptive engine control strategies, in the search for solutions to the problem of emissions constrained minimisation of fuel consumption. The model would be used to represent an automotive system typical of a mid-range saloon car; such that simulated fuel and emissions data could be recorded continuously as the 'vehicle' is 'run' over a particular driving schedule.

Whatever the reasons for the simulation study, it is usual to decompose the automotive system into smaller units, in order to simplify the modelling task. Conveniently we can consider the system as five main modules: vehicle, transmission, engine, controller, and driver. In an advanced vehicle all five modules will be present as in figure 4.1 Here the driver reacts to feedback from the vehicle, engine and other external stimuli; and manipulates the vehicle via the controller. Most road vehicles can in fact be considered using the same five modules; the differences being largely in the module interconnections and the exact tasks of the controller.

This chapter describes an approach to the modelling of automotive systems, particularly relevant to the treatment of engine control system optimisation. As such it will be evident that the level of detail necessary needs to be greater for some modules than for others. Figure 4.2 illustrates the information flow typical of the type of the system being addressed.

4.2 THE VEHICLE MODULE.

Fundamental to this module is the calculation of the road load retarding force; and the tractive force generated at the tyre-surface interface. It follows that if the tractive force $F_w(t)$ exceeds the retarding force $F_r(t)$ then a corresponding acceleration of the vehicle will occur; and a deceleration if the retarding force is the greater.

An equation of the form

$$F_r(t) = C_R Mg + \tfrac{1}{2} A_f \rho C_D V(t)^2 + Mg \sin(\alpha(t)) + F_B(t) \qquad (4.1)$$

may be used if the effects of rolling resistance, aerodynamic resistance and road gradient, need to be identified uniquely (see section 3.2.2). Often, however, a simpler polynomial representation of empirical data is sufficient yielding an equation of the form

$$F_r(t) = a_0 + a_1 V(t) + a_2 V(t)^2 + F_B(t) \qquad (4.2)$$

where $a_0 \ldots a_2$ are coefficients; $V(t)$ is the vehicle velocity and $F_b(t)$ is the force due to the application of the service brakes.

The exact form of the equation will depend on the simulation aims; and if, for instance, the effect of tyre pressures on vehicle performance was to be predicted, then the pressure would have to be incorporated into the rolling resistance portion of the road load expression.

The net tractive force $(F_w(t) - F_r(t))$ gives the vehicle acceleration

$$\dot{V}(t) = (F_w(t) - F_r(t))/M \qquad ms^{-2} \qquad (4.3)$$

where M is the mass of the vehicle, and $F_w(t)$ is the tractive force at the tyre-road interface.

4.3 THE TRANSMISSION MODULE.

The module boundary definitions are somewhat arbitrary: the clutch is seen here as part of the transmission rather than the engine to which it is attached. The transmission thus includes all the components 'between' the engine flywheel and the road surface.

Each component in the transmission or drive-train generally has a rotational inertia associated with it; which can be lumped with other inertias if they are 'rigidly' coupled. When inertias are joined by a compliant shaft then the individual dynamics need to be considered, as for the simple example of figure 4.3. Here the dynamics are modelled by application of Newton's laws.

$$J\dot{\omega}_2(t) = \tau_2(t) - \tau_1(\theta(t),\dot{\theta}(t))$$
$$\dot{\theta}(t) = \omega_1(t) - \omega_2(t)$$

where J is the inertia of the element; $\tau_1(\theta(t),\dot{\theta}(t))$ the reactive torque due to deflection $\theta(t)$ and damping (proportional to $\dot{\theta}(t)$) in the coupling; $\tau_2(t)$ is an applied torque, and $\omega_1(t)$, $\omega_2(t)$ are angular velocities.

It will be clear from this that the order of the final transmission model will depend on the number of significant rotating inertias and torsional elements. For many studies such as fuel economy optimisation, the drive-train resonances may be of secondary importance, and a simpler non-compliant drive-train model can be adopted.

Assuming a non-compliant drive-train representation, the power influencing the vehicle due to the net tractive force, equates to the change of energy of the rotating components and of the inertial mass of the vehicle.

$$(F_w(t) - F_r(t)).V(t) = \sum_{k=1}^{n} J_k \omega_k(t)\dot{\omega}_k(t) + MV(t)\dot{V}(t) \qquad (4.4)$$

Also since the angular velocity of the kth rotational element can be referred to the vehicle speed V by the relation

$$\omega_k(t) = r_k(t)V(t) \qquad (4.5)$$

where r_k is the speed ratio corresponding to the element, we have

$$(F_w(t) - F_r(t)).V(t) =$$
$$\sum_{k=1}^{n} J_k r_k(\dot{r}_k(t).V(t) + r_k(t)\dot{V}(t)).V(t) + MV(t)\dot{V}(t) \qquad (4.6)$$

which in the case of a constant transmission ratio, yields

$$(F_w(t) - F_r(t)) = \left[\sum_{k=1}^{n} J_k r_k^2 + M\right].\dot{V}(t) \quad \text{N} \qquad (4.7)$$

The rotational elements can thus be viewed as contributing to the effective inertial mass of the vehicle; and for a fixed ratio transmission the acceleration of the vehicle in gear g_i

$$\dot{V}(t) = (F_w(t) - F_r(t))/M_j(g_i) \qquad ms^{-2} \tag{4.8}$$

where the tractive force $F_w(t)$ is calculated assuming no effect due to rotational inertias (cf. equation 4.3).

The clutch output speed $\omega_o(t)$ (for gear g_i) is related to the vehicle speed V(t) by the equation

$$\omega_o(g_i,t) = \frac{V(t).R(g_i).R_a}{r_w} \qquad s^{-1} \tag{4.9}$$

for a non compliant drive-train. Here $R(g_i)$, R_a are the gear ratio and axle ratio respectively; and r_w is the rolling radius (m) of the driving wheels.

The efficiency of the gearbox (for a given gear), and final drive unit depends both on the torque transmitted and on the input speed. The total torque loss for a given input speed can be viewed as the sum of the churning loss (at zero output torque) and a load dependant torque loss. Often the torque loss is measured at a high output torque; a reasonable estimate of the torque loss at lower load conditions may then be obtained by linear interpolation between this and the churning torque.

It is in fact more sensible to use a representation of the torque loss, rather than refer to the gearbox or final drive 'efficiency'. In simple transmission models, however, a constant efficiency may be used for a particular gear; or efficiency may include some dependency on input speed. In the case of a constant gearbox efficiency $\eta(g_i)$ and axle efficiency η_a being used, the torque required at the output of the clutch T_{co} to give a torque of T_w at the wheels, is

$$
\begin{aligned}
T_{co} &= \frac{T_w}{R(g_i).R_a.\eta(g_i).\eta_a}, \quad T_w \geqslant 0 \\
&= \frac{T_w.\eta(g_i).\eta_a}{R(g_i).R_a}, \quad T_w < 0
\end{aligned}
\tag{4.10}
$$

Efficiencies are awkward to handle satisfactorily under conditions approaching zero load, and the above equation shows two distinct relationships depending upon the direction of power transfer; both these problems are best avoided by explicit use of the transmission torque, or power loss, rather than the concept of efficiency.

The Clutch

The clutch, normally located on the engine flywheel of a vehicle fitted with a manual transmission, allows the driver to regulate the torque in the gearbox input shaft. In particular it permits the engine speed to exceed that of the input shaft when the lower speed would stall the engine, or when the engine could not supply sufficient torque at the lower speed.

Detailed modelling of the clutch would require consideration of the compliance and damping designed into the device, and perhaps also the complex effects due to the temperature of the friction material. The concept of the clutch as a torque limited device is fundemental and is illustrated by figure 4.4. Here the clutch characteristic is given by

$$T_L = (T_{max} - T_0)u + T_0 \qquad (4.11)$$

where the torque transmitted by the clutch cannot exceed the limit T_L, which depends on some control input u having range [0,1]. This control is likely to be related to the pressure clamping the clutch faces together, and at its maximum (u=1) the clutch could transmit a torque of T_{max} - usually designed to be considerably in excess of the maximum engine torque. At the opposite extreme of operation corresponding to a driver 'disengaging' the clutch (u=0), the clutch can transmit only a very small torque T_0 - insufficient to overcome frictional forces for a stationary vehicle.

4.4 THE ENGINE MODULE.

4.4.1 Modelling Approaches.

An automotive engine is exceedingly complex, and presents a formidable task for the modeller hoping to represent its processes in fine detail. Once again, the level of detail to be incorporated into the model depends on the simulation tasks to be performed; the desired accuracy of the results, and the engine information available or measurable given the practical constraints.

Engine models exist in a number of forms, but may be classed as one of three types:

a) the detailed combustion models (4.1-4.3) requiring many computations per combustion cycle.

b) the discrete form (4.4) which requires a single computation per engine cycle.

c) the continuous system model that is not synchronised with the engine combustion cycle.

The approach developed below is of the latter, asynchronous, type; and is applicable to the control problems addressed by this study. Until now most automotive powertrain simulations have been based entirely on static engine data, and hence have not been able to deal with the dynamics of actuators, exhaust after-treatment, or cold-start phenomena. This means that the use of such models in investigation of the validity of various control algorithms is severely limited. Probably the most comprehensive published simulation of this type to date is that of Beachley and Frank (4.5). This model is based on steady-state data from a <u>pre-calibrated</u> engine; though not suitable for engine control studies, it is useful for investigating the effect of vehicular or drive train modifications on vehicle fuel consumption. Such modifications are typically: gear ratios, aerodynamic drag, continuously variable transmissions (CVT) and energy storage devices (e.g. flywheel). The development given below, while using steady-state engine data, incorporates certain dynamic effects; in particular the flywheel dynamics, the temperature dynamics and their effect on emissions, and the basic behaviour of the fueling system.

It will be readily appreciated that if the engine data set is sufficiently over-determined, steady-state fuel and emission flows can easily be predicted for any speed, load and control setting within the range of the data: a measure of the over-determination of a particular datum point can be estimated by examining the effect on the regression if it is removed from the set of data. (A minimal effect will be observed if the point is sufficiently over-determined). This assumes that the well known technique of multiple linear regression has been used to model the steady-state engine characteristics, as is often done. Having determined the steady-state flows it should be possible to estimate the transient flows, according to known dynamic trends. In order to reasonably reflect these trends it is essential that a clear indentification is made of the major factors involved in emission production.

Fundamental to this modelling approach is the concept that in a reciprocating engine the combustion during one cycle is completed and isolated from the next cycle. This implies that if all the parameters influencing combustion in the cylinder are identical, then there is no difference between the steady state engine operation and the transient one. Thus steady-state data from the engine test-bed can be utilised, and the transient simulation facilitated by rapid alteration of operating conditions.

The major difficulty arises in predicting the operating condition unless exact characterisation of the engine has been undertaken with particular regard to thermal inertia of various parts of the engine, actuator dynamics, and vehicle dynamics. For certain simulation work it will not be necessary to have an exact representation of a particular car; merely a qualitative representation of the trends influencing emission production. In the case of the control studies reported in a later chapter, it is reasonable to expect a strategy which 'works' on an 'exaggerated' model to provide useful results on the real vehicle, as experience has shown that control system design procedures are tolerant of small modelling errors.

4.4.2 Steady State Characterisation.

The Data Base

For any engine model such as this it is necessary to have detailed information about the performance characteristics of the engine. Basic data consist of such variables as pollutant flows, fuel flow and system temperatures, expressed in terms of the engine operating conditions.

Useful data cannot be accumulated without the expenditure of considerable amounts of time and money. It is therefore essential that a systematic and careful approach is adopted in recording and screening the engine data to ensure the integrity of the data base, and its suitability for the use to which it will be put.

Computerised methods of data collection have come to the fore (4.6-4.9) greatly easing the difficulty in setting up an engine operating point, and allowing vast quantities of data to be accumulated in a shorter time. As has been indicated, the collection of data is only part of the problem; attention must be focussed also on data processing methods which assist the screening of the raw engine data during and after acquisition. This problem is summarised by Mencik and Blumberg (4.8):

> "Those data entries having excessive error that could lead to spurious emissions and fuel economy projections must be identified and eliminated. Another important function of such data processing methods is the smoothing of the engine performance data in order to remove random experimental scatter before the data are used ... it is desirable to interpolate experimental data for combinations of the independant variables that at a given speed/load combination, were not directly measured."

For engine calibration or control purposes it will be necessary to record data over a wide range of engine speeds and loads. Spark advance (SA), air fuel ratio (AF) and possibly exhaust gas recirculation need to be adjusted to characterise the engine over this region. The latter control variable (EGR) is not immediately relevant to the European problem as it stands, and its exclusion from the simulation greatly simplifies the modelling, as strong interaction occurs between EGR, AF and inlet manifold pressure. Apart from these interactions the effect on otherwise clearly identifiable trends is unknown, but likely to be considerable.

Apart from the basic requirement of good sensors and data collection equipment, the control of ambient conditions in the engine test cell minimises many of the problems that have dogged engine testers for years. A further safeguard is the interactive examination of data: the use of sophisticated data analysis software can assist in rapid isolation and elimination of suspect data, using versatile graphical display facilities. It is well known that the human brain can rapidly recognise patterns that may be difficult or laborious to express in mathematical terms. The visual data check is therefore a powerful tool in the screening procedure. Even if an automatic procedure is adopted, the trained operator can monitor the operation, checking 'stray' points and initialising a re-test. One big advantage is that data can be examined during engine testing, thus obviating the need to re-install a particular engine in the test cell at a later date - a costly and now unnecessary burden on a facility which is often fully utilised as it is.

Regression Modelling of the Engine Data Base

To enable the model to be constructed the implicit characteristics of the data base need to be expressed in a functional algebraic form. Particularly useful are the widely known principles of multiple linear least-squares regression (4.10-4.12), which have been described with reference to automotive data, by Mencik and Blumberg (4.8).

In order to express the engine 'flows' as functions of the chosen control variables, the underlying relationship between the response (dependant) variable Y and the predictors X is held to be the functional form

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \ldots + \beta_k X_k + \varepsilon$$

where ε is the error of estimation.

In regression an estimate $\hat{Y}$ of Y is obtained by determining the regression coefficients b_i of

$$\hat{Y} = b_0 + b_1 X + b_2 X + \ldots + b_k X_k \qquad (4.12)$$

in order to minimise ε .

The least-squares technique chooses the coefficients that minimise the 'residual sum of squares'

$$\sum_{j=1}^{n}(Y_j - \hat{Y}_j)^2$$

over the n samples of the data set (Appendix A).

$\hat{Y}$ is seen to be a linear combination of the predictors $X_1,\ldots,X_k$; which in this application are functions of the engine variables (e.g. SA,SA*SA,RPM,AF*AF*SA) and as such are correlated (multicollinear). This means that each b_i does not give an indication of the unique contribution of the X_i predictor to the regression, but also measures some of the effect of the other predictors on Y.

The engine characteristics which can be represented in this way include steady state torque, temperatures, induction mass air flow, fuel flow and emission flows.

In the formation of the engine emission models a transformation can be made in the representation of the dependant variable, such that the logarithm of the response variable is regressed instead of the variable itself. The prediction is then obtained from the model by exponentiation. The three engine emissions of concern are carbon monoxide (CO), unburnt hydrocarbons (HC) and oxides of nitrogen (NOx); all of which have a range of several orders of magnitude. It is difficult with the normal linear representation to avoid physically meaningless negative values being predicted. The logarithmic transformation eliminates such a problem allowing also the representation of a function involving exponential dependance using relatively few terms in the regression. Another valuable effect from a physical stand-point is the way in which the transformation tends to keep the relative estimation error dY/Y constant over the range of X, rather than the absolute error dY; the latter being the case in the more usual linear representation of the response variable.

4.4.3 Rotational Dynamics.

Often it is possible to represent the inertias of the rotating and reciprocating parts of the engine as a single inertia effective at the flywheel. In this case the dynamics can be written

$$J_F \dot{\omega}_F(T) = T_E(u(t), \omega_F(t)) - T_{c1}(t) \tag{4.13}$$

where the acceleration of the inertial mass of the flywheel $\dot{\omega}_F(t)$ depends on the inertia J_F, and the torques acting on it.

Here the engine torque T_E is a function of the engine controls and states, and the torque on the clutch input T_{c1} depends on the remainder of the powertrain.

4.4.4 Temperature Dynamics and Effects on Emissions.

While it may be reasonable to ignore warmed-up engine temperature dynamics in the prediction of vehicle fuel economy; it is far more difficult to predict the cumulative emissions of a vehicle driven over a schedule, if temperature transients are disregarded. In section 3.3 it was noted that cylinder surface temperature and exhaust manifold surface temperature, in particular, have an effect on certain exhaust emissions. Steady state emission flow data is obtained during the mapping procedure, and includes the steady state engine temperatures implicitly as a function of the engine operating point. When the engine is driven to a new operating point the temperatures 'lag', giving rise to a different emission flow than is predicted by the steady state map, if the particular engine emission is sensitive to temperature. For example: going from steady operation at a high-speed high-load condition to a low-speed low-load condition will produce lower hydrocarbon emissions at the low-load point than predicted by the steady state data; this is due to the elevated temperature in the cylinder (thinner quench layer) and exhaust manifold.

Carbon Monoxide

Variations of cylinder and exhaust manifold surface temperatures were shown by Huls et al (4.13) and Myers et al (4.14) to have negligible effect on CO emissions; the only significant engine operating variable being air-fuel ratio.

Unburnt Hydrocarbons

Hydrocarbon emissions are difficult to predict accurately. They depend on cylinder surface area, mixture preparation, air-fuel ratio, ignition advance and also on the cylinder surface temperature, exhaust manifold temperature and residence time. Myers and Alkidas (4.14) and Huls et al (4.13) illustrate the hydrocarbon trends with respect to the temperatures; and it is clear that cylinder surface temperature does not influence oxidation in the hot exhaust manifold but does affect the emission of hydrocarbons from the cylinder.

Emission production in combustion engines is complex (4.15, 4.16), so for simulation purposes some relatively simple means of incorporating the effects of cylinder and exhaust manifold surface temperatures is desirable.

Analysis has shown (4.13, 4.14, 4.17) that essentially in the range $14 < AF < 20$ the sensitivity of hydrocarbons to cylinder surface temperature is independant of air fuel ratio, and for a given operating point the hydrocarbon concentration can be expressed

$$[HC] = a + kT \qquad (4.14)$$

where a and k are constants and T is the absolute temperature. At leaner mixtures (AFR>20) there is a greater sensitivity due to other factors; and the relationship is generally nonlinear.

Speed appears to have little effect on the sensitivity, but at much lower temperatures there is an increase, particularly for very lean mixtures.

The presence of excess oxygen in the hot exhaust manifold causes oxidation of hydrocarbon species and a resultant reduction in emissions (4.13, 4.14, 4.18). The reaction has been described by the Arrhenius reaction rate equation (4.13, 4.19):

$$[\dot{HC}] = k[HC][O_2]T^{\frac{1}{2}}\exp[-E/RT] \tag{4.15}$$

where E is the activation energy

R is the molar gas constant

This equation broadly states that the amount of HC reacting is proportional to the excess oxygen and strongly related to the exhaust temperature.

Nitrogen Oxides

Oxides of nitrogen are mainly formed during the compression stroke (4.15, 4.16) by dissociation of air in the high flame temperatures. Post-flame temperatures drop rapidly in the expansion stroke inhibiting the decomposition of nitric oxide to the levels predicted by the equilibrium equations, because of the slower rate of reaction. The lower concentration of oxygen in the gases at this time is also thought to be contributory to the 'frozen equilibrium'. Additionally no effect has been observed due to the cooling of the exhaust manifold (4.14).

The fact that NOx production is very sensitive to peak combustion temperatures leads one to expect there to be some influence due to the cylinder surface temperature at the beginning of the compression stroke. A significant effect has been noticed (4.13, 4.14) particularly at weaker mixtures: an effect that is virtually independant of engine speed. The relation can be described as (ref 4.14):

$$\log (NOx) = kT + c \tag{4.16}$$

where k and c are constants corresponding to a particular air-fuel ratio; T is absolute temperature.

Given the steady-state concentration NOx_1 at temperature T_1 it is then possible to predict the concentration NOx_2 at the actual temperature T_2 by

$$\log(NOx_2) = k(T_1 - T_2) + \log(NOx_1) \tag{4.17}$$

if we have knowledge of the gradient k.

Induction Manifold Temperature

The importance of a model for inlet manifold temperature relates to the effect it has on the transient performance of the fueling system: and in as much as the air-fuel ratio affects torque and emissions, then induction manifold temperature dynamics must be considered if vehicle emissions performance is to be simulated.

The surface temperature of the inlet manifold can depend on a number of factors. For a coolant heated manifold the temperature will usually be close to that of the coolant under steady state idle conditions; while at full power the flow of fuel and air causes a considerable lowering of the surface temperature (4.20). Additionally many induction systems control the inlet air temperature by taking a proportion of the flow from the region of the hot exhaust manifold.

The steady state temperature may be modelled by regression analysis techniques as indicated in section 4.4.2. However, under cold start conditions for instance, the manifold temperature will reach an equilibrium temperature below the steady state value. For the water heated manifold, this equilibrium temperature may be reasonably predicted, using the linear relationship

$$T_i = T_{is}.(1 - C(T_{cs} - T_c)/T_{cs}) \qquad (4.18)$$

where T_i is the equilibrium manifold temperature

T_{is} is the steady-state manifold temperature

T_c is the coolant temperature

T_{cs} is the steady-state coolant temperature

C is a constant determined empirically.

The dynamics may be represented as a first order system

$$\dot{\theta}_i(t) = (\theta_{ie}(t) - \theta_i(t))/\tau \tag{4.19}$$

where θ_{ie} is the equilibrium temperature

θ_i is the dynamic temperature

and τ is the time constant

Cylinder and Exhaust Manifold Surface Temperatures

The steady state values for these temperatures may be represented by a regression model, and their dynamics as above (eq. 4.19).

Coolant Temperatures

As indicated above, the coolant temperature requires simulation in order to determine the cylinder surface temperature, and because of the effect the coolant may have on the induction manifold. It is reasonable to use a linear first or second order model and to assume that the final temperature is controlled adequately by the thermostat. The 'warm-up time' may well be found to be affected by the work done by the engine, and also by the flow of cooling air which is proportional to vehicle velocity.

In many vehicles the exhaust manifold has been used to provide a 'hot spot' to assist in cold start conditions. Often no direct heating is applied but the temperature of the induction manifold is sensed and the inlet air temperature is controlled by using a proportion of the air from the vicinity of the hot exhaust manifold.

4.4.5 Modelling the Fueling System.

Carburettor and induction system dynamics are very complex and require considerable effort in order to construct a fully validated model. The mathematical model, however simple or complex, must reflect the most important features of the real physical system, as far as they have been determined. The fueling system is designed to provide (generally) a homogeneous mixture of known air-fuel ratio under steady engine conditions. However, under dynamic conditions the air-fuel ratio in the combustion chambers is known to differ markedly from the set point.

Typically a lean excursion is evident for increasing throttle angle, and a rich excursion for decreasing throttle angle; the magnitude of this phenomena being inversely proportional to the fuel and inlet manifold temperatures.

The lean/rich behaviour of induction dynamics can be attributed principally to the formation of a liquid fuel film in the manifold. Many factors influence film thickness, evaporation, and transportation of the fuel within the manifold (4.21, 4.22); and in a detailed model it may be necessary to take into account the pulsative air flow, and the variation of mixture strength from cylinder to cylinder (4.23). A considerable amount of work has been directed towards improving the fueling rate compensation (4.24-4.25): models used in this work necessarily include fuel film, and manifold filling phenomena in order to address the needs of both speed-density and mass-flow fuel metering strategies (4.26-4.27).

The dominant characteristics of the carburation and induction dynamics can be represented by the transfer function model illustrated in figure 4.5 The physical system can be considered as having a fast flow and a slow flow (4.4): the fast flow including air, evaporated fuel and fuel droplets; the slow flow being a representation of the liquid fuel film. The air circuit in this model has negligible lag, while the fuel circuit is considerably slower being dominated by the fuel lag. The controller, which may be mechanical or electronic, allows adjustment of the nominal air-fuel ratio; which under steady-state conditions, is the actual ratio.

Figure 4.5 also shows a small controller lag and a fuel enrichment device: this latter may be an accelerator pump, which is a feedforward element intended to reduce the lean excursion that results from an increasing throttle angle.

Stivender (4.28) used a similar transfer function model to show that alternative fueling strategy (Engine Air Control EAC) can obviate the need for such enrichment devices. This EAC system meters air with reference to a demanded fuel flow; thus allowing the incorporation of a lag in the faster (air) circuit to compensate the slower (fuel) dynamics.

The throttle pump in this model is represented by

$$\begin{aligned} \dot{m}_p(t) &= \min[K_p\dot{\phi}(t), \dot{m}_{p_{max}}] \;, \quad 0 \leq \phi(t) \leq \phi_p \\ &= 0 \;, \quad \dot{\phi}(t) \leq 0 \end{aligned} \tag{4.20}$$

where $\dot{m}_p(t)$ = pump mass fuel flow

$\phi(t)$ = throttle angle

$\dot{m}_{p_{max}}$ = maximum pump mass flow

K_p is a constant, and the pump has no output for throttle angles greater than ϕ_p.

The controller time constant τ_c is fixed as a result of the system design, but the slow fuel lag depends on the manifold temperature as well as the design. Given that the manifold temperature is being modelled, one way of representing the effect on the fuel lag τ_f is the linear function

$$\tau_f(T_i(t)) = \tau_{f_0} - (\tau_{f_0} - \tau_{f_{100}}).T_i(t)/100 \qquad (4.21)$$

where τ_{f_0} is the value of τ_f at 0°C

$\tau_{f_{100}}$ is the value of τ_f at 100°C

and T_i is the temperature of the inlet manifold (°C)

4.4.6 Driveability.

Driveability is a term used to describe some subjective phenomenon or phenomena related to the performance of a vehicle powertrain. A vehicle with good driveability delivers its power to the road in a smooth and dependable manner irrespective of the road or weather conditions. In design or analysis of vehicle systems with a view to addressing driveability problems, it is necessary to construct a model which incorporates the compliant elements of the drivetrain.

In a narrower sense 'driveability' can be used to refer to the 'smoothness' of the engine power delivery; in which case a non-compliant model can still be used to give some indication or comparison of say two different engine calibrations. The full effect on the vehicle will not be simulated, however; neither will the effect of transmission component behaviour (e.g. sudden clutch take-up).

Various attempts have been made to define and quantify what are really subjective phenomena associated with unwanted variations in engine output torque (4.29-4.30). Everett (4.29) describes driveability in terms of idle quality, stalls, hesitation, sag or stumble, backfire, surge and stretchiness: all these terms describe partial or complete loss of power manifesting itself in various ways. Generally these problems can be attributed to inadequate actuator transient response, or steady-state combustion instability.

Driveability is effectively a constraint in the minimum fuel problem: it is not possible to minimise fuel consumption and NOx while simultaneously achieving 'best' driveability. The essentially weak mixture required for the former greatly degrades the latter; and the resultant engine calibration generally has to be richer than might otherwise be required. The transient operation of a carburetted engine results in significant differences between the desired and the actual fuel quantity inducted into the cylinders. The lean excursions, exacerbated by a cold manifold and uneven fuel distribution between cylinders (4.31) results in worse hesitation, surge, stumble and even stalling.

Matsumoto et al (4.32) used engine output torque fluctuations as a measure of surge-type driveability in their simulation work. Though drive-train interactions do give rise to driveability phenomena, cylinder pressure variations are fundamental (4.33). Fukushima et al (4.34) related the variation of indicated mean effective pressure (IMEP) with the results from Everett's vehicle mounted accelerometer (4.29), and obtained remarkable agreement. McFarland and Wood (4.35) developed an analog computer which continuously measured IMEP (actually the integral of PdV), which Dohner (4.36) used to measure combustion variability for driveability evaluation. The problem still remains of isolating the combustion stability component from 'normal' trends in cylinder pressure due to changes in throttle angle. This at present has only been done by using a prescribed driving schedule and performing a reference run with the engine controls set for 'minimum variance'; whereupon all successive runs are compared with the reference.

A comparison of accelerometer and cylinder pressure measurement techniques was illustrated by Dohner (figure 4.6 from reference 4.36). Such data could be used in the formation of a regression model for engine roughness type driveability.

Engine knock, though not strictly concerned with driveability, is an abnormal combustion problem which limits engine performance and puts constraints on the engine controls (4.37). Ignition advance is one of the major factors influencing engine knock (see summary in table 4.1 from reference 4.19) and the general trends are shown in figure 4.7

TABLE 4.1

Summary of Factors Influencing Tendency for Engine Knock
(from reference 4.19)

FACTOR	EFFECT	PRO-KNOCK	ANTI-KNOCK
Compression ratio	Increase	*	
Ignition timing	Advance	*	
Throttle setting	Full open	*	
	Partially open		*
Mixture strength	Weak		*
	Slightly rich	*	
	Over rich		*
Engine speed	Increase		*
Inlet air temperature	Increase	*	
Inlet air pressure	Decrease		*
Cooling water temperature	Increase	*	

4.5 THE CONTROLLER MODULE.

The role of the controller in an advanced powertrain as shown in figure 4.1, is to provide an interface between the driver and the system; ensuring that the system responds to the demands of the driver in a predictable manner. Its task may be to interpret the driver's pedal position and to control gear ratio, clutch position and throttle angle, in order to achieve a desired acceleration or cruising speed; complex system interactions and strict driveability, safety and performance criteria can easily make this an impossible control problem for the driver alone to cope with.

In a conventional non-automatic powertrain system with an S.I. engine, the driver controls the transmission components, the throttle angle and the service brakes. The controller is then often viewed as the electro-mechanical or electronic subsystem, which adjusts the air-fuel ratio and/or the ignition spark advance according to sensed engine speed and load; this is illustrated in figure 4.2, and such functions are termed 'engine management' or 'engine control'. Modern electronic engine controls allow arbitrarily complex functions to be incorporated to control spark advance (4.38); and similarly complex calibrations for nominal air-fuel ratio (for electronic carburettors or fuel injection equipment).

The accurate control of ignition advance is relatively easy compared to the task of controlling actual air-fuel ratio. The simulation of these systems can readily be undertaken by representing the spark advance and nominal air-fuel ratio by the actual functions of the engine variables used in practice. These functions will be relatively simple for the mechanical systems fitted to the majority of mass produced engines; but the more advanced controllers (4.39-4.40), which may be adaptive in nature, are not necessarily more difficult to simulate as the algorithm will probably have been developed for implementation in a microcomputer.

4.6 THE DRIVER MODULE.

4.6.1 Introduction.

An automotive simulation attempts to represent the complicated functions associated with a real or prospective vehicle by means of the simplest acceptable approximation. A necessary part of this is the modelling of driver behaviour. The extent to which realism is incorporated in the driver model depends on the aims of the simulation, as the task of characterisation is not easy.

Frequently only simple functions of throttle angle are required, such as step or ramp changes. Additionally 'crowds' or constant manifold vacuum accelerations are required, and often an open loop control is sufficient. The tasks of the driver varies from the control of accelerator and brake pedals in the automatic or advanced vehicle (figure 4.1), to the control of throttle angle, clutch position and gear ratio in the conventional manually operated vehicle (figure 4.2). It is this latter case which is considered in detail below.

It will be appreciated that no two drivers perform in the same manner given identical cars, road conditions and route. A driver will interact with the vehicle and his performance will depend on his ability, his mood, urgency and the prevailing conditions with regard to traffic and weather. These facts are recognised and form a central feature of the excellent work done by Nowottny and Hardman (4.41-4.42) for example.

Their studies yielded no less than seven parameters associated with driver behaviour, and the concept of driver desired speed profiles was introduced. The desired speed profile is governed by route features such as speed limits, bends, hills and stop points. On a certain section of the route the desired speed may be 60 m.p.h., the national speed limit: the driver determines how he wishes to achieve that target speed i.e. acceleration rate, gear usage, etc. This is illustrated in figure 4.8 and should be compared with the fixed speed-time profile of the ECE15 cycle (figure 3.4), which is independent of the characteristics of the car or driver, and thus inhibits many interactions present in the real system.

The driver model adopted by Nowottny (4.42) characterised the desired acceleration as

$$\dot{V}_d = k_n(V_d - V) + k_c \quad , \quad V_d > V$$

where V_d is the desired speed

V is the actual speed

and k_m, k_c are 'driver determination' parameters

when $V_d < V$ then the desired deceleration is give by

$$\dot{V} = k_d(V_d - V)$$

with k_d increasing with decreasing velocity as V_d tends to zero, to avoid overshooting the desired stop point.

The driver is further characterised by the way in which he utilises the engine; it being noted that drivers tend not to use the whole of the available operating range of the engine. He tends to avoid labouring the engine, uses higher engine speeds during overtaking or hill climbing, and generally avoids the use of full throttle at any engine speed. Nowottny uses three parameters associated with this behaviour and a further parameter which describes the minimum time taken for a gear change.

4.6.2 Driving Schedule Tracking.

If it is necessary to simulate true driver behaviour including anticipation (in order to examine driver related performance effects on fuel economy, emissions or driveability) then the above approach provides a good basis. However if the desire is to follow a given speed-time profile (which represents the interactions of a particular driver-vehicle combination on a particular occasion) then the above approach is unsuitable. In the context of this present work it was decided that the simulated vehicle should be able to follow a driving schedule with reasonable accuracy; the term 'desired speed' will thus hereinafter refer to the speed prescribed by the driving schedule for that particular instant.

It is evident that some form of closed loop controller is necessary for the task. A simple linear feedback controller of the form shown in figure 4.9 could be implemented to represent the driver. The feedback path represents the driver's perception of vehicle speed and acceleration which he compares with his desired speed. The forward path relates to his response in adjusting the throttle to minimise the error. Due to the integrating action the velocity error will be zero under cruising conditions. The simplicity of this controller is attractive but it has weaknesses which limit its usefulness in this application. A major problem exists in choosing the controller gains to give adequate performance over a wide range of operating conditions. It is possible to devise a means of scheduling these according to vehicle speed, load and gear selected; but any change then made to the system would degrade the 'driver' performance.

An actual driver learns to adapt his response when presented with a new vehicle; and it would be desirable to incorporate some learning behaviour in the driver model. The gains would then be adjusted in response to both operating conditions and system changes by correlating system behaviour with throttle action. The alternative and simpler implementation used in this work assumes the driver has full knowledge of the system; he is thus able to specify precisely the required throttle angle to achieve his desired acceleration.

4.6.3 Driver 'Desired Conditions'.

If the driving schedule specifies the desired vehicle velocity V at times t_1, t_2, t_3,... (usually at intervals of 1 second) then the desired acceleration to reach the next cycle velocity

$$\dot{V}_d(t) = \frac{V_d(t_i) - V(t)}{t_i - t} \tag{4.22}$$

where V(t) is the vehicle velocity at the current time t, and $V(t_i)$ is the velocity specified by the driving schedule at time $t_i > t$.

The torque required at the roadwheels for this acceleration in gear g_i is

$$T_{dw}(g_i) = (M_j(g_i).\dot{V}_d(t) + F_r(t)).r_w \tag{4.23}$$

where F_r is the road load force due to aerodynamic and rolling resistance (see section 4.2); and r_w is the rolling radius of the wheel. $M_j(g_i)$ the effective vehicle mass in gear g_i (see section

4.3) is related to the actual vehicle mass M by the equation

$$M_j(g_i) = M(1 + \gamma(g_i)) \qquad (4.24)$$

where $\gamma(g_i)$ is the fractional increase in effective vehicle mass for gear g_i; thus taking into account the effect due to the reciprocating and rotating parts of the powertrain. The gear-dependancy of the relationship is of course due to the fact that the engine has a greater angular acceleration when in a low gear than when a high gear is selected. Table 4.2 shows that the vehicle can seem almost 50% heavier when moving off from rest.

The engine speed in gear g_i is obtained from the vehicle speed V(t)

$$\omega_e(g_i,t) = \frac{V(t).R(g_i).R_a}{r_w} \qquad (4.25)$$

where $R(g_i)$ and R_a are the gear and rear axle ratios respectively, and r_w is the tyre rolling radius (m).

Clearly to calculate the desired torque at the engine flywheel ($T_{DE}(g_i,t)$ we must know the gearbox efficiency in each gear $\eta(g_i)$ and the rear axle efficiency η_a; giving

$$T_{DE}(g_i,t) = \frac{T_{DW}(g_i,t)}{R(g_i)R_a\,\eta(g_i)\eta_a}\ , \quad T_{DW}(g_i,t) \geqslant 0 \qquad (4.26)$$

$$T_{DE}(g_i,t) = \frac{T_{DW}(g_i,t).\eta(g_i)\eta_a}{R(g_i).R_a}\ , \quad T_{DW}(g_i,t) < 0 \qquad (4.27)$$

for the driving and overrun conditions respectively. A more detailed powertrain simulation would probably need efficiency coefficients that are themselves functions of speed and torque (4.5), or a representation of the gear and speed dependant torque loss of the gearbox and final drive unit.

TABLE 4.2

Effect of rotating and reciprocating powertrain components on effective vehicle mass for a typical mid-range saloon car.

Gear [g_i]	$\frac{\text{Effective Vehicle Mass}}{\text{Actual Vehicle Mass}}$ [$1 + \lambda(g_i)$]
Neutral	1.06
1	1.48
2	1.21
3	1.12
4	1.09

4.6.4 A Gear Selection Algorithm.

The purpose of the module is to select the gear, throttle angle, clutch position and brake setting to obtain the desired torque at the wheels ($T_{pw}(g_i,t)$). If this torque is large and negative then the vehicle brakes may be required; if positive and large then the engine may not be able to supply the torque. Ordinarily, however, only moderate torques are required; which can be supplied by the engine. In this case the driver really only has to choose the gear, as there will exist only one throttle angle yielding the required torque for a particular gear. Using an alternative gear effectively causes the engine to operate at a different speed and a different torque. Fuel economy, noise and vehicle response depend heavily on these latter conditions and must somehow be incorporated into the gear selection process.

The gear selection algorithm for a car fitted with an automatic transmission responds to vehicle speed and load (inlet manifold vacuum). While a similar algorithm may be appropriate for a simulation using a representation of a manual gearbox, it can only be effectively used when the engine has a fixed calibration: a calibration change alters the engine torque characteristic, necessitating a modification to the automatic gear change algorithm.

A reasonable gear change philosophy would be to select the highest gear that is easily able to supply the necessary torque within the engine speed range. This can be used to bias operation towards the low engine speed and high torque region of good fuel economy, and is not too distinct from the behaviour of an actual driver. It is now necessary to consider how this underlying gear selection philosophy can be implemented in the driver module.

Having calculated the engine speed and required engine torque for each gear, as detailed above, the highest gear within the engine speed range is chosen as a trial gear. This gear may be used if it can provide a sufficient excess of torque above that currently desired at the engine, i.e. if

$$T_{max}(g_i).K_{gc} > T_{oe}(g_i) \tag{4.28}$$

where $T_{max}(g_i)$ is the maximum available engine torque at the speed applicable to gear g_i, and the gear-change constant K_{gc} is less than unity - a value of 0.6 may be suitable.

If the test of equation 4.28 did not include the gear-change constant then too high a gear would often be chosen; resulting in poor vehicle response to acceleration demands, and the need for frequent re-selection of gear ratio.

When 4.28 is not satisfied the next lower gear needs to be examined for suitability in a similar manner. The higher gear may be selected if the lower gear would result in an excessively high engine speed; or if neither can supply the required torque but the higher gear has a torque advantage, i.e.

$$T_{max}(g_i) - T_{DE}(g_i) \leqslant T_{max}(g_{i-1}) - T_{DE}(g_{i-1}) \qquad (4.29)$$

This implies that the lower gear would operate at an engine speed that is higher than that which gives peak torque.

Should the new trial gear be unable to supply the torque and the higher gear offers no advantage, then the lower gears must be tested in the same way, until the 'best' gear has been determined.

First gear is chosen if the engine speed required for higher gears is below the minimum engine speed, or if itself is best able to meet the torque requirements. If the vehicle speed and the desired vehicle speed are zero then conditions must be set for idle, with transmission neutral selected. Normally, however, desired speed and torque will be non-zero.

At this point the clutch is considered: as the vehicle powertrain is being modelled as a non-compliant system, the clutch dynamics may be ignored. In particular, no clutch slip need be considered except in the lowest gear, when it needs to be determined with regard to the permitted engine speed range, and desired engine torque (i.e. the engine speed would be too low without clutch slip, or insufficient torque can be provided by the engine unless its speed is increased). The clutch is said to be slipping when the speed ratio (clutch output speed/clutch input speed) is less than unity.

Having determined the most suitable gear and, if necessary, the clutch speed ratio, the algorithm needs to check if the current gear is the same. If this is so no gear change is required and the correct vehicle brake setting and throttle angle need to be determined. Braking will be required if the predicted engine torque at closed throttle is greater than the desired engine torque in the chosen gear; the predicted engine torque being obtained from the engine steady state torque model at closed throttle. The braking force

$$F_B = \frac{T_{min}(g_i) - T_{de}(g_i).R(g_i).R_a \eta(g_i)\eta_a}{r_w} \qquad (4.30)$$

where $T_{min}(g_i)$ is the minimum available engine brake torque in the chosen gear g_i.

If braking is required this implies that the throttle is fully closed, otherwise the braking force

$$F_B = 0$$

and the throttle setting needs to be chosen to give the desired engine torque.

Selection of Clutch Speed Ratio

Clutch slip is adjusted when first gear is chosen. It is necessary to choose the clutch speed ratio to give either the desired torque at the transmission primary shaft, or if this is not possible, the maximum available within the engine speed range.

The clutch speed ratio σ lies in the range [0, 1] and is defined as the ratio of the clutch output to clutch input speed; hence unity implies no slipping. The clutch has a torque limited characteristic described by figure 4.4, and equation 4.11. If the clutch is controlled such that the torque limit is below the torque produced by the engine then the engine accelerates accordingly, reducing the speed ratio; if the torque limit exceeds the torque produced by the engine then no slip occurs and the engine is considered to be connected directly to the transmission.

Under conditions of clutch slip a certain amount of energy will be dissipated in heat. This results in an effective torque loss in the device which increases from zero with decreasing speed ratio. Some representation of this may be made using a linear equation similar to (4.11), ie.

$$T_{co} = T_{\varepsilon}(C_o + (1 - C_o)\sigma) \qquad (4.31)$$

where T_{ε} and T_{co} are the torques on the engine and gearbox sides of the clutch respectively, and C_o is the torque conversion ratio (T_{co}/T_{ε}) at a speed ratio σ of zero.

The algorithmn used here attempts to select a clutch speed ratio that is as near unity as possible consistent with the engine speed range and desired clutch output torque. If the vehicle speed is very low and deceleration is desired, then the clutch is disengaged in readiness for a stop.

If the clutch output speed is above the minimum (idle) engine speed then a test speed ratio of unity is initially chosen; otherwise σ is chosen consistent with the minimum engine speed, i.e.

$$\sigma = 1 \quad , \omega_o \geq \omega_{E_{min}}$$
$$\sigma = \frac{\omega_o}{\omega_{E_{min}}} \quad , \omega_o < \omega_{E_{min}} \qquad (4.32)$$

where $\omega_{E_{min}}$ is the minimum engine speed and ω_o the clutch output speed.

Having an initial speed ratio, the maximum available engine torque is determined, and the resultant clutch output torque calculated, from (4.31). If this is able to meet the desired output torque then the new desired engine torque is

$$T_{DE}(t) = \frac{T_{DE}(g_i,t)}{(C_o + (1 - C_o)\sigma)} \qquad (4.33)$$

where $T_{DE}(t)$ is the new desired engine torque, and $T_{DE}(g_i,t)$ is the previous desired torque (now the desired torque on the clutch output shaft).

If the torque requirement would not be met at this trial value of speed ratio, available clutch output torque is checked for higher engine speeds (increasing clutch slip). At a certain engine speed peak torque is produced; but as the engine speed is allowed to increase by increasing clutch slip, a smaller proportion of the torque is transmitted by the clutch. If the desired clutch output torque could not be attained then the speed ratio giving the highest output torque is most suitable.

Using the speed ratio chosen the new desired torque at the engine is determined from (4.33). A pseudocode version of this algorithm is shown in figure 4.10.

Gear Changing

If there is a difference between the current gear and the choice arrived at by the algorithm above, then a gear change needs to be performed. The driver module signals the condition for clutch disengaged and gearbox neutral selected, and reduces the throttle angle to its idle setting. In practice gear changes take a finite time; thus this condition must exist for the 'gear change period', before re-selection of a new gear. With the gear in neutral the force required to meet the vehicle acceleration requirements is calculated, giving the braking force

$$F_B(t) = \min[M_i(g_o).\dot{V}_d + F_R(t),\ 0.0] \qquad (4.34)$$

where $M_i(g_o)$ is the effective vehicle inertial mass in neutral. This means that the brakes are not applied unless a deceleration greater than that provided by the road load F_R is required.

Characteristically drivers do not change gear at frequent intervals: it is a trivial matter to incorporate into the gear changing algorithm, a limit on the frequency. A minimum of about 4 seconds between changes is usually suitable, which can be overridden in the event of engine overspeed or underspeed.

4.6.5 Driver Control of Throttle Angle.

In the above description the assumption was made that the throttle angle could be determined, which would give a specific steady-state torque in order to meet the vehicle acceleration requirement. Given that there exists a model for the steady state engine torque as some function involving engine speed, throttle angle, air-fuel ratio and spark advance; then it is possible to determine the required throttle angle. A driver, however, has physical and mechanical constraints on his ability to deliver any desired throttle angle: the simplest model of his behaviour is probably a slew limit, such that the angle is not permitted to change at too high a rate, i.e.

$$\dot{\phi}_{min} \leq \dot{\phi}(t) \leq \dot{\phi}_{max}$$

where $\dot{\phi}_{min}, \dot{\phi}_{max}$ are the minimum and maximum limits on the time rate of change of throttle angle $\phi(t)$.

It is not desirable to model the throttle angle in a similar manner to the torque function, because the inconsistencies between the two models would result in poor driving schedule tracking. Instead it is better to use the torque model itself, and determine the throttle angle by solving

$$f(\phi) = T_E(\omega_E, SA, AF, \phi) - T_{DE}(g_i) = 0 \qquad (4.35)$$

where engine speed (ω_E), air-fuel ratio (AF), spark advance (SA) and desired engine torque ($T_{DE}(g_i)$) are constant.

The torque is an increasing monotonic function in throttle angle (figure 4.11) hence there exists only one solution to 4.35; which may be obtained by, for example, the method of bisection.

If we have two points ϕ = a and b where $f(\phi)$ has opposite signs, the continuity of $f(\phi)$ implies that it must have at least one zero between them: and in this case only one zero (otherwise an increase in throttle angle may result in a decrease in torque). The bisection method relies on sign changes to detect a zero. If $f(a).f(b) < 0$ we evaluate $f(\phi)$ at the mid-point $m = (a+b)/2$. There are three possibilities:

i)	$f(m) = 0$	;	m is the zero
ii)	$f(a).f(m) < 0$	;	the zero is in [m,a]
iii)	$f(b).f(m) < 0$	;	the zero is in [b,m]

This brackets the zero in an interval half the length of the original. The procedure is then repeated until we arrive at a sufficiently small interval containing the zero.

The bisection method is useful as it is extremely easy to implement, always converges and is stable with respect to limiting precision: this latter means that as we attempt to obtain very small values of the computed function, such that the finite word length of the computer enters into the process; the bisection method does not give absurd approximations. The bracketing property itself has the advantage that it allows easy and reliable determination of convergence.

The stopping criterion can be the number of iterations, or some check on convergence such as successive function values or the size of the interval. In the present task a difference in function value of less than 1Nm may be considered satisfactory convergence.

The major draw-back of the bisection method is its relatively slow convergence rate compared to say, Newton-Raphson or secant methods. However in this application the guaranteed convergence of the bisection method, and the fact that it requires only a single function evaluation for each iteration are major factors in the choice. The rather slack convergence criterion ensures that the region of limiting precision is avoided and the solution is found with few iterations. If it is desirable to obtain a more accurate solution, then the bisection method can be combined with another method (e.g. the secant method) as the solution is approached (4.43).

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NOTATION

V_d	desired vehicle velocity
V	actual vehicle velocity
F_r	vehicle road load force
C_R	rolling resistance coefficient
M	vehicle mass
g	acceleration due to gravity
A_f	vehicle frontal area
C_D	vehicle aerodynamic coefficient
ρ	air density
α	grade angle
F_w	tractive force
$a_0 \ldots a_k$	coefficients used in rolling road expression
F_b	braking force on vehicle
J	inertia of system element
ω_1	angular velocity of system element
ω_2	angular velocity of system element
τ_1	reactive torque of compliant element
τ_2	torque applied to system element
θ	angular deflection in compliant element
J_k	inertia of system element k
ω_k	angular velocity of element k
r_k	speed ratio for element k
$M_j(g_i)$	inertial mass of vehicle in gear g_i
$\omega_o(g_i,t)$	clutch output speed for gear g_i

$R(g_i)$	ratio of gear g_i
R_a	final drive ratio
r_w	wheel rolling radius
T_L	clutch limiting torque
T_{max}	maximum clutch torque capacity
T_0	limiting torque of 'disengaged' clutch
u	control input
$\eta(g_i)$	efficiency of gear g_i
η_a	final drive efficiency
T_{co}	torque at clutch output
T_w	torque at road wheels
Y	response variable
$\beta_0 \dots \beta_k$	model coefficients
$X_1 \dots X_k$	model regressors
ε	error of estimation
$\hat{Y}$	estimate of Y
$b_0 \dots b_k$	coefficients produced by regression
J_f	effective engine flywheel inertia
ω_f	engine speed
T_E	engine net torque
T_{ci}	torque at clutch input
[HC]	hydrocarbon concentration
a	parameter
k	parameter
T	absolute temperature
$[O_2]$	oxygen concentration
E	activation energy

R	molar gas constant
T_i	equilibrium inlet manifold temperature
T_{is}	steady state inlet manifold temperature
T_c	coolant temperature
T_{cs}	steady state coolant temperature
θ_{ie}	equilibrium temperature
θ_1	dynamic temperature
τ	time constant
$\dot{m}_p$	throttle pump mass fuel flow
K_p	throttle pump constant
$\dot{m}_{pmax}$	upper limit on $\dot{m}_p$
ϕ	throttle angle
ϕ_p	throttle pump stroke limit
τ_c	fuel controller time constant
τ_f	effective time constant of fuel circuit
τ_{f0}	value of τ_f at 0°C
τ_{f100}	value of τ_f at 100°C
k_m	parameter of driver model
k_c	parameter of driver model
k_d	parameter of driver model
T_{Dw}	torque desired at the road wheels
$\gamma(g_i)$	effective fractional increase in vehicle mass due to inertial effects
ω_e	engine speed
T_{DE}	desired engine torque
T_{Max}	maximum available engine torque
K_{gc}	gear change constant
T_{min}	minimum available engine torque

σ	clutch speed ratio
T_{co}	clutch output torque
C_e	clutch torque conversion ratio (disengaged)
$\omega_{E_{min}}$	minimum (idle) engine speed
ω_o	clutch output speed
$\dot{\phi}_{min}$	throttle angle lower slew limit
$\dot{\phi}_{max}$	throttle angle upper slew limit

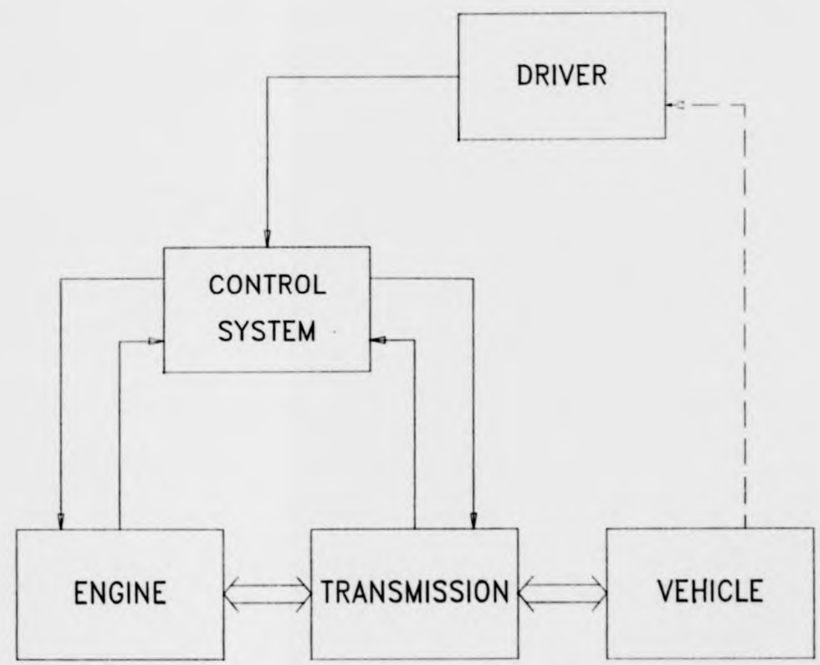


Figure 4.1 Schematic of an Advanced Automotive Vehicle

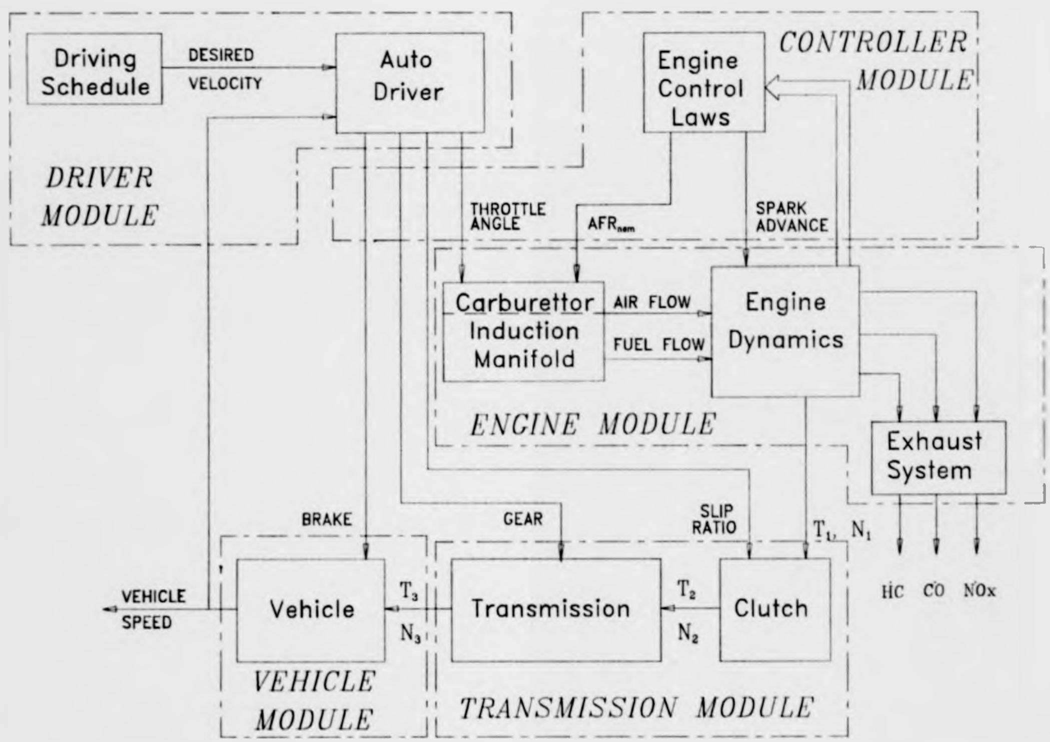


Figure 4.2 Simulation Model – Simplified Information Flow

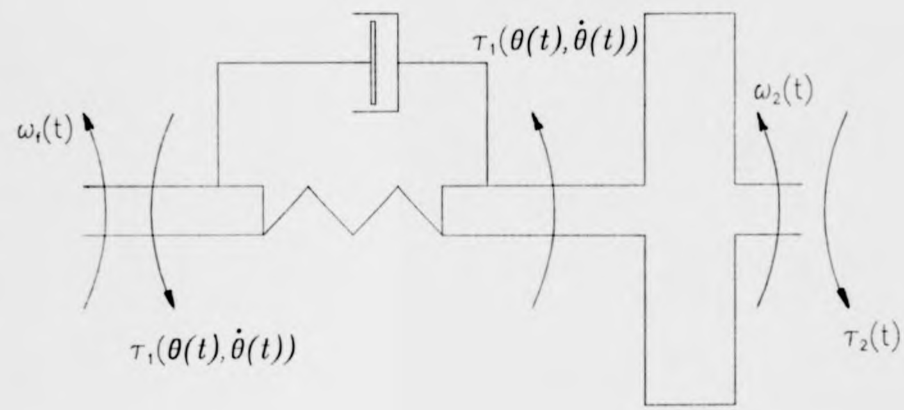


Figure 4.3 Lumped parameter model of a rotating mass on a compliant shaft

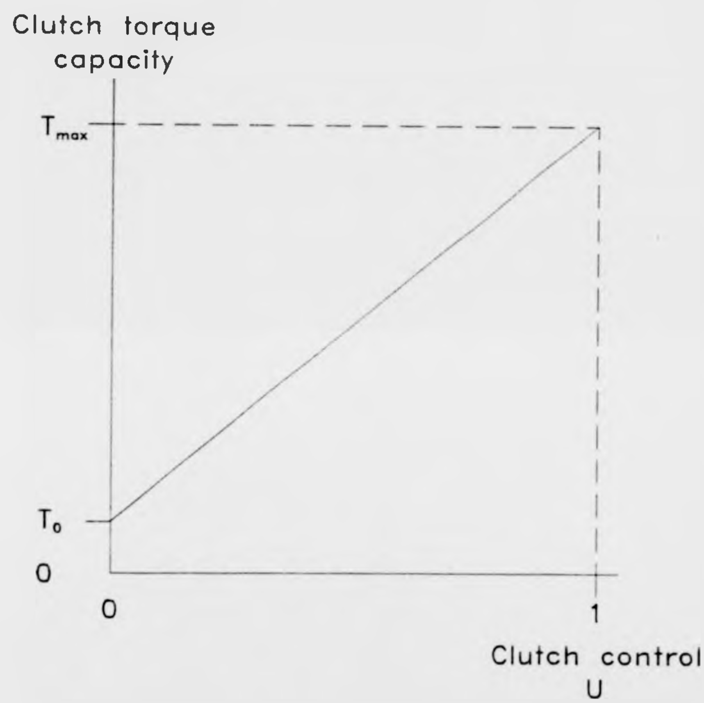
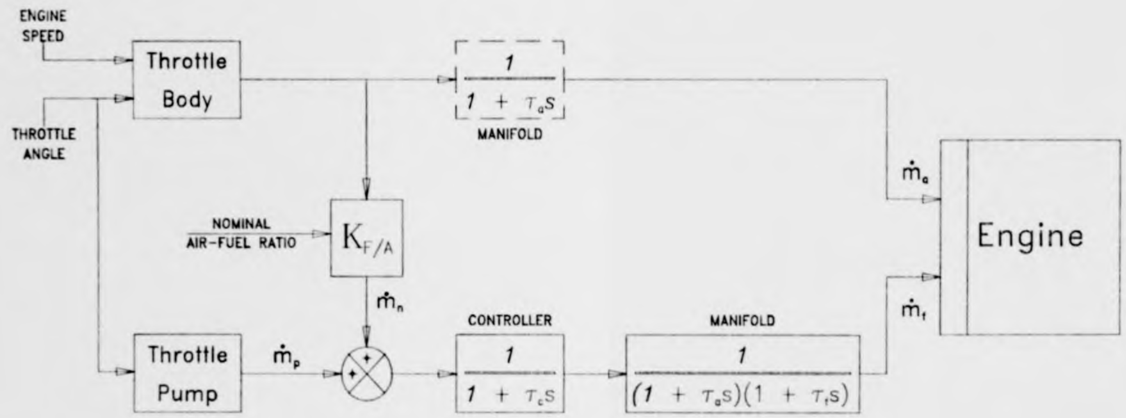


Figure 4.4 Clutch Characteristics



Symbols: τ_c controller lag (s)
τ_f fuel lag in manifold (s)
τ_a air lag in manifold (s)
$\dot{m}_n$ nominal fuel flow
$\dot{m}_p$ throttle pump flow
$\dot{m}_f$ inducted fuel flow
$\dot{m}_a$ inducted air flow

Figure 4.5 Symbolic representation of the Induction Model

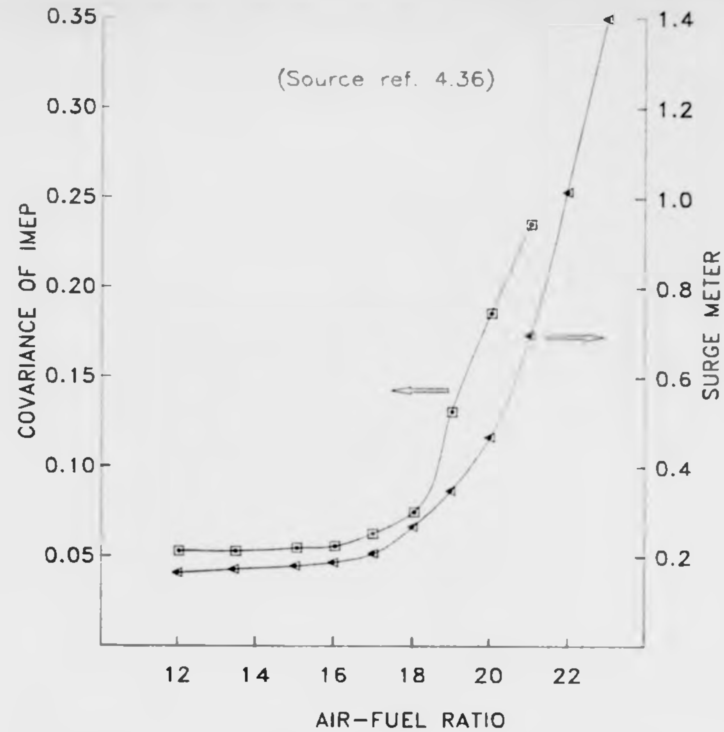


Figure 4.6 Covariance of IMEP & Surge versus Air-Fuel Ratio

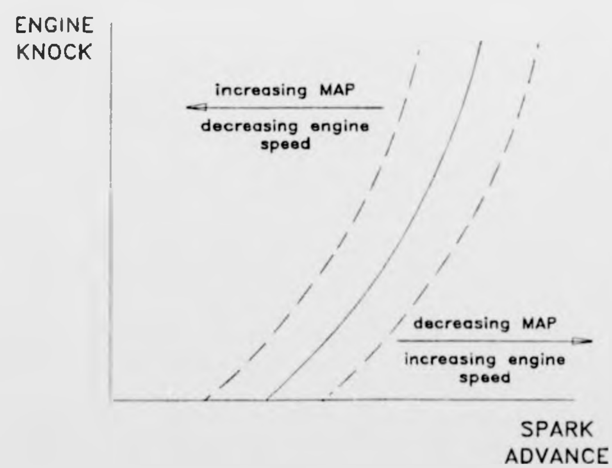


Figure 4.7 Engine Knock Trends with Ignition Timing

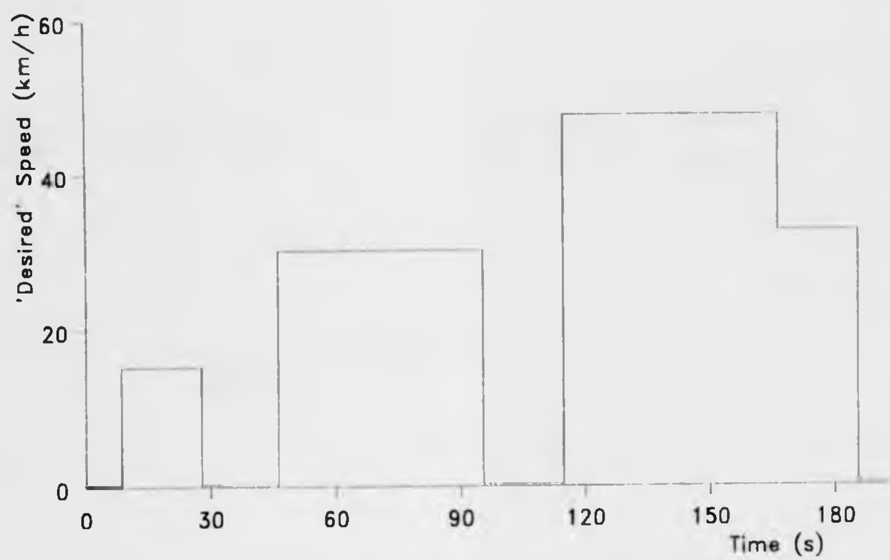


Figure 4.8 Driver 'desired' speed for ECE-15 schedule

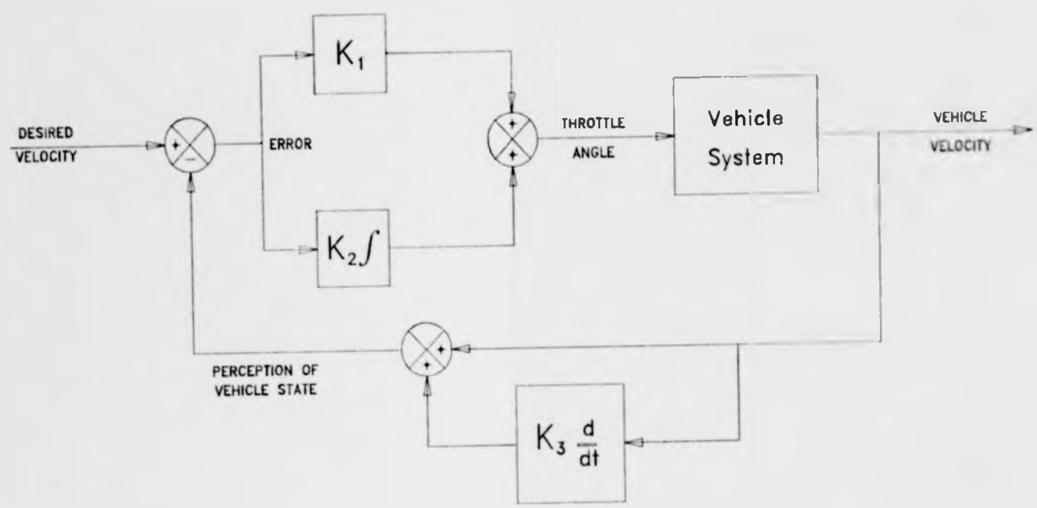


Figure 4.9 Simple Automatic Driver for Schedule Following

```
START

gear = 1 (only for first gear)

speed ratio σ = 1     (clutch fully engaged condition)

DO WHILE engine speed is too low

      reduce speed ratio σ

END DO

IF available torque at the clutch output too low THEN

      DO WHILE a reduction in σ increases output torque
                without engine overspeed

           reduce speed ratio

      END DO

ELSE IF vehicle speed very low and deceleration required THEN

      speed ratio σ = 0   (clutch disengaged condition)

END IF

FINISH  (speed ratio σ chosen)
```

Figure 4.10: Pseudocode version of algorithm for selecting clutch speed ratio

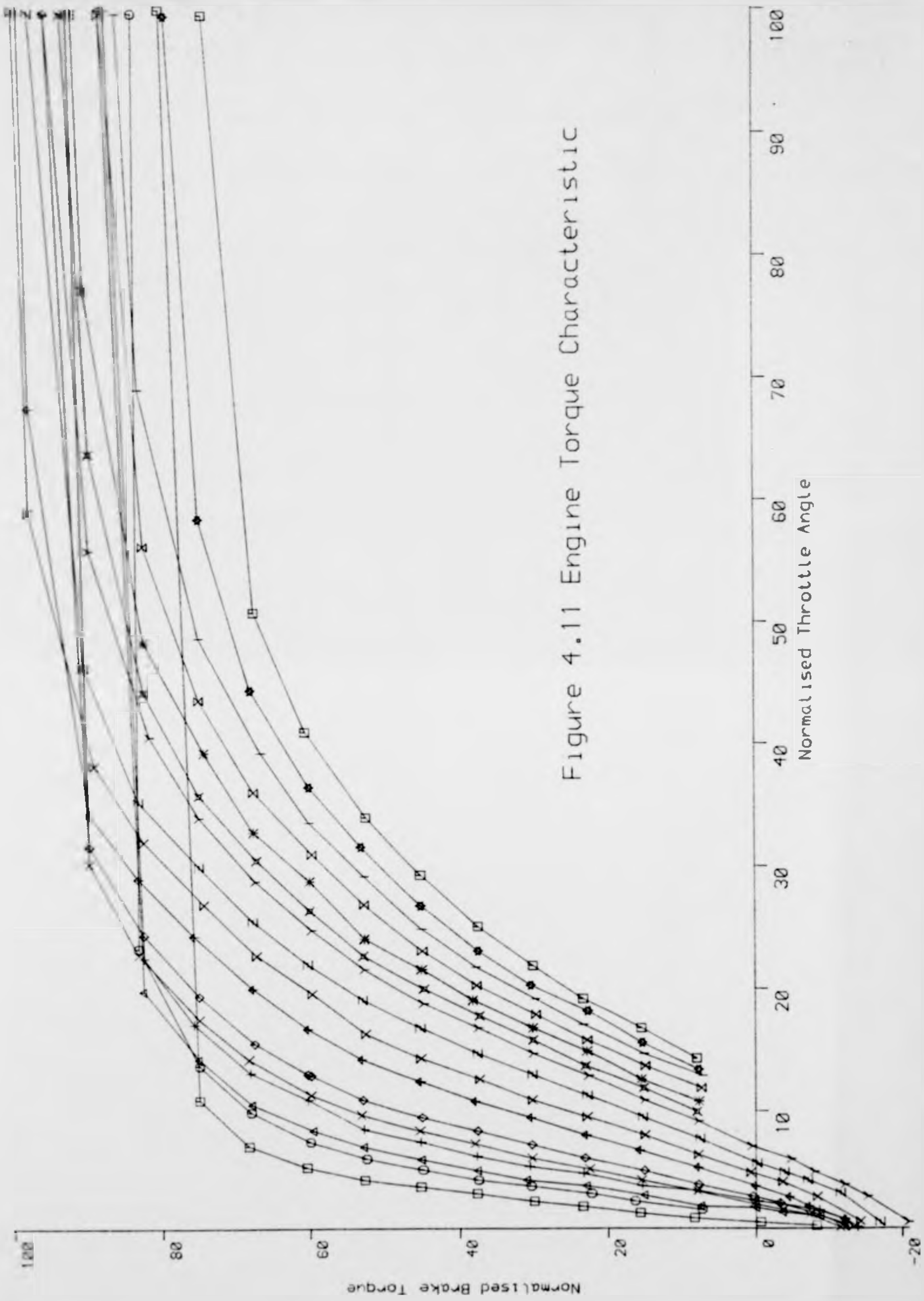


Figure 4.11 Engine Torque Characteristic

CHAPTER 5

DEVELOPMENT OF A CONTINUOUS AUTOMOTIVE SYSTEM SIMULATION

5.1 INTRODUCTION.

This chapter describes the development of a simulation facility based on the modelling described in chapter 4. The Continuous Automotive System Simulation (CASS) is coded in FORTRAN and designed specifically for evaluation of certain engine control problems, as described in section 4.1

The nonlinear dynamic model is a closed loop characterisation, formulated in a modular manner, recognising the physical structure of the system. It is a continuous system model that is not synchronised with the engine combustion cycle. This approach differs from the discrete form (5.1) which requires one computation per engine cycle; and the detailed combustion models (5.2, 5.3) which require many computations per cycle.

Separate subroutine modules are used to represent individual vehicle subsystems such as the fueling system. The automatic driver is responsible for adjusting driver controls (gear, clutch, throttle) within the program loop, while the nominal engine controls (air-fuel ratio, spark advance) are specified as functions of the engine operating condition.

A simplified representation of the information flow in the model is shown in figure 4.2. It will be appreciated that the automatic driver requires knowledge of the whole system in order to generate the optimum control settings (gear, clutch slip, throttle). This compares with the driver of an actual vehicle who will adjust the throttle and the 'throttle control gain' according to the gear selected, the acceleration required and the throttle response of the vehicle. He in turn will also have to concern himself with gear selection on the basis of the engine torque characteristic and speed range, balancing the throttle and the clutch as he does so. These complex interactions add greatly to the modelling task.

5.2 STAGES IN MODEL DEVELOPMENT.

It is useful to consider the development of the Continuous Automotive System Simulation model in the following stages:

1. Obtain static mapping data for a suitable engine over a wide range of engine operating conditions. These data consist of fuel flow, emission flows and relevant engine states recorded on magnetic tape or disc. Often such data is collected from computer assisted testing of an engine on a special test-bed.

2. Regression of the engine data to obtain steady state models of certain engine variables.

3. Code the equations obtained in (2) above, to form part of the simulation program.

4. Model drive-train and vehicle characteristics from engine and vehicle specifications. The resultant mathematical expressions are coded for inclusion in the simulation, and thus enable the desired engine operating point to be computed.

5. Develop suitable algorithms for driver behaviour to allow automatic selection of gear, throttle and clutch position.

6. Formulate a suitable carburation and induction model.

7. Having coded the simulation modules, combine the automatic driver and other dynamic models in a program loop to permit integration of the system variables (fuel rate, emissions flow, coolant temperature, etc.)

8. Provide modules for the input of program constants and output of the run-time data.

9. Thoroughly test the model behaviour to ensure reasonable correspondence with actual vehicles.

5.3 THE ENGINE DATA BASE.

Engine data utilised in this research program was kindly provided by the Ford Motor Co. Ltd. A 1600 cc. displacement engine was used and data was recorded over a wide range of engine speeds and loads. Spark advance (SA), air-fuel ratio (AF) and exhaust gas recirculation (EGR) were adjusted to characterise the engine over this region. The latter control variable (EGR) was not required for the current model, as it is not immediately relevant to the European problem as it stands. The elimination of EGR from the simulation greatly reduces the modelling difficulties as strong interaction occurs between EGR, AF and inlet manifold pressure. Its inclusion would have seriously impaired the validity of the simulation, as apart from these interactions the effect on otherwise clearly identifiable trends is unknown, but likely to be considerable; necessitating an extensive additional planned program of experiments.

Considerable effort is expended by the Ford Motor Company to achieve consistent high quality in their engine data. This includes computer assisted screening of the data as referred to in section 4.4.2; a procedure which was repeated using the Interactive Data Analysis (IDA) facility at the University of Warwick, after the data tapes had been decoded and the selected information retrieved. The resulting data base was then analysed using multiple linear least-squares techniques (see section 4.4.2 and Appendix A).

5.4 STEADY STATE CHARACTERISATION.

5.4.1 Regression Analysis Procedure.

In this work the predictors were corrected for their means by transforming equation 4.11 into

$$Y = b_0 + b_1(X_1 - \overline{X}_1) + b_2(X_2 - \overline{X}_2) + \ldots + b_k(X_k - \overline{X}_k) \qquad (5.1)$$

This affects only the b_0 coefficient but reduces the numerical rounding errors in the computer.

The regression of engine flows was carried out in three or four independent engine variables. A full fourth order polynomial regression using four engine variables requires seventy coefficients to be determined, including a constant. As many of these coefficients are statistically insignificant, a semi-automatic technique was used to reduce the number of terms. This 'backward elimination' of redundant coefficients has the benefit of improving the over-determination (5.4). The computerised selection of coefficients to be retained was monitored by interactive examination of the resultant model: the computer program automatically selects the most significant coefficients based on a t-test criterion of statistical relevance. The non-significant coefficients in the regression are restrained a few at a time and the resultant regression examined against the original data to confirm its efficacy.

NAG routine G02BAF was used to compute the basic statistics for input to the multiple regression routine G02CBF. The t-statistic for the result was calculated in the normal way and saved for subsequent examination and analysis (figure 5.1). Another NAG routine G02CJF was used initially as it had the advantage of operating on raw data, outputted the t-statistic directly, and performed the regression with much greater speed than G02BAF/G02BCF. Problems occurred due to the short word length used in the compilation of the routine, but this was eventually solved by obtaining the source program from NAG. It was not however the end of the problems: though the regression coefficients produced were identical with G02BAF, which was used subsequently, the t-statistic proved inconsistent. The Numerical Algorithms Group affirmed that there was a 'bug' associated with the variance/covariance matrix, and it was decided to abandon G02CJF in favour of the much slower G02BAF.

Each regressed variable was expressed in terms of engine speed, spark advance, air-fuel ratio and throttle angle or torque. The engine output torque model is unique in that it is a regressed variable as well as being used in the regression of other variables. In each model either torque or throttle angle would be used dependant upon the quality of fit of the data points. Careful testing of each model was necessary to ascertain the behaviour if extrapolation beyond the range of the data base was attempted.

5.4.2 Engine Flow Models Based on Regression.

The following variables were modelled for steady-state conditions using the regression procedure outlined above:

Engine output torque
Induction system mass air flow
Carbon monoxide emission
Nitrogen oxides emission

Unburnt hydrocarbon emission

Cylinder surface temperature

Exhaust manifold temperature

Induction manifold absolute pressure

Engine knock level

Engine hesitation (lean-limit of combustion)

A single subroutine for calculating engine flows could incorporate all the regression coefficients for the above variables in a two-dimensional array. The coefficient set, for the required variable to be evaluated, would be selected using an argument in the subroutine call. However this approach was not adopted as all 70 terms in the models would be evaluated, many of which have zero coefficients.

It was more in keeping with the modular structure of the program to code each function as a separate routine. Many of the variables need evaluating during each loop of the main program, and it was therefore vital to avoid the time consuming evaluation of the zero terms in the polynomial.

A function subprogram was made for each of the variables and the terms in the regression polynomial were grouped to form a single FORTRAN statement, minimising the number of multiplications. This latter approach has a speed advantage though does not offer the flexibility possible if all the regression terms are calculated. This is no drawback in this application but would be inconvenient if more than one set of engine data were to be used; as this would then necessitate changes to the function subprograms as well as to the input data.

Coefficients for all the functions were kept in a common block for initialising by the data input routine INICON.

TABLE 5.1

Predictors used in the regression polynomials.

(SA = spark advance; RPM = engine speed; AF = air-fuel ratio;
T = engine torque or throttle angle)

Term	Components	Term	Components
1	CONSTANT	36	SA**4
2	SA	37	AF**4
3	AF	38	RPM**4
4	RPM	39	T**4
5	T	40	SA**3 * AF
6	SA**2	41	SA**3 * RPM
7	AF**2	42	SA**3 * T
8	RPM**2	43	AF**3 * SA
9	T**2	44	AF**3 * RPM
10	SA * AF	45	AF**3 * T
11	SA * RPM	46	RPM**3 * SA
12	SA * T	47	RPM**3 * AF
13	AF * RPM	48	RPM**3 * T
14	AF * T	49	T**3 * SA
15	RPM * T	50	T**3 * AF
16	SA**3	51	T**3 * RPM
17	AF**3	52	SA**2 * AF**2
18	RPM**3	53	SA**2 * RPM**2
19	T**3	54	SA**2 * T**2
20	SA**2 * AF	55	AF**2 * RPM**2
22	SA**2 * T	57	RPM**2 * T**2
24	AF**2 * RPM	59	SA**2 * AF * T
25	AF**2 * T	60	SA**2 * RPM * T
26	RPM**2 * SA	61	AF**2 * SA * RPM
27	RPM**2 * AF	62	AF**2 * SA * T
28	RPM**2 * T	63	AF**2 * RPM * T
29	T**2 * SA	64	RPM**2 * SA * AF
30	T**2 * AF	65	RPM**2 * SA * T
31	T**2 * RPM	66	RPM**2 * AF * T
32	SA * AF * RPM	67	T**2 * SA * AF
33	SA * AF * T	68	T**2 * SA * RPM
34	SA * RPM * T	69	T**2 * AF * RPM
35	AF * RPM * T	70	SA * AF * RPM * T

Engine Output Torque.

Three function subprograms were developed for the torque model: FNCTQ, AMAXTQ, AMINTQ. FNCTQ predicts the engine brake torque given the engine speed, spark advance, air-fuel ratio and throttle angle. The terms in the regression polynomial are:

1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 15, 18, 19, 22, 26, 31, 55

from Table 5.1.

AMAXTQ and AMINTQ functions return the estimate of brake torque at maximum and minimum throttle angles respectively. As such they do not require any predictors involving throttle angle. The coefficients were obtained from those used by FNCTQ.

Intake Mass Air Flow.

The function subprogram FNCAIR uses 14 terms in the regression polynomial:

1, 4, 5, 8, 9, 14, 15, 18, 28, 35, 38, 48, 51, 56

from Table 5.1. It returns an estimate of mass air flow as a function of engine speed, ignition advance, air-fuel ratio and throttle angle.

Carbon Monoxide Emission.

The regression polynomial used in the function subprogram FNCCO gives the natural logarithm of the carbon monoxide mass flow rate. The following predictors from Table 5.1 are used:

1, 2, 3, 4, 5, 8, 13, 14, 17, 25, 26, 27, 34, 35, 37, 43, 45, 50, 64, 65, 70

Unburnt Hydrocarbon Emissions

The function subprogram FNCHC provides the natural logarithm of hydrocarbon flow for steady-state engine operation. The polynomial uses the following predictors from Table 5.1:

1, 2, 3, 4, 5, 8, 10, 13, 14, 15, 19, 20, 21, 22, 26, 27, 28, 29, 32, 35, 37, 44, 45, 48, 49

For the simulation a look-up table was used to provide an estimate of the percentage of hydrocarbons oxidised at a particular temperature and air-fuel ratio. These figures were based on the results published in References 5.5-5.9. It must be borne in mind that the intention was to incorporate the trend and not attempt an exact representation of the reaction for this particular engine; which would not have identical exhaust manifold volume or residence time.

Subroutine HCADJ takes the steady-state (regressed) hydrocarbon flow and adjusts it according to the deviation of cylinder surface and manifold temperatures from the steady-state conditions (see section 4.4.4).

Nitrogen Oxides Emissions.

FNCNOX is a function subprogram which returns the natural logarithm of nitrogen oxides flow under steady-state conditions. The following predictors (from Table 5.1) were used in the model

1, 2, 3, 4, 5, 7, 9, 13, 14, 15, 18, 22, 39, 31, 37, 38, 39, 54, 56, 65, 70

Subroutine NOXADJ adjusts the steady-state flow to accommodate the trend of NOx emissions with deviations from steady-state cylinder surface temperature. Reasonable estimates of the constants were obtained from the cited literature for various air-fuel ratios, and linear interpolation used for intermediate points (see section 4.4.4).

Cylinder Surface Temperature.

The function subprogram FNCCYL returns the steady-state mean cylinder surface temperature using the following predictors from Table 5.1:

1, 2, 3, 4, 5, 6, 7, 8, 10, 15, 18, 46, 48, 55, 69

The dynamic temperature is estimated using a small first-order lag (section 5.5.7).

Exhaust Manifold Temperature.

Steady-state exhaust manifold temperature is provided by the function subprogram FNCEX. The following predictors were used (from Table 5.1):

1, 2, 3, 4, 5, 6, 7, 8, 15, 18, 46, 48, 65

As deviations from steady-state exhaust temperatures can affect hydrocarbon emissions a first-order lag was used to simulate the dynamic temperature (section 5.5.7).

Induction Manifold Temperature.

The engine used for this simulation work had a water heated inlet manifold. At steady-state idle the manifold temperature was the same as that of the coolant, as is usually the case (5.10). At full power the flow of fuel and air caused a depression of 40°C below coolant temperature in steady-state. The steady-state characteristic is defined by the function subroutine FNCINL which uses the following predictors from Table 5.1:

1, 3, 4, 5, 7, 9, 14, 17, 19, 38

When the coolant temperature is below normal (i.e. during warm-up) the manifold reaches an equilibrium temperature below the steady-state value. A linear relationship between coolant temperature and equilibrium temperature (as expressed in equation 4.18) was thought to be suitable for this simulation, with a value of about 0.75 for the constant C.

The dynamics of the manifold temperature were simulated using a short first-order lag (section 6.4.7).

Induction Manifold Absolute Pressure.

Inlet manifold pressure is often used in engines as a control variable, as it is almost linearly proportional to engine torque (load). A function subprogram FNCMAP was used to provide this variable and the following predictors (Table 5.1) were used:

1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 15, 19, 26, 29

Driveability

A simple model for engine roughness was devised, representing hesitation and surge type torque behaviour (see figure 5.2). This type of phenomena was discussed in section 4.4.6: putting it into the context of the driveability of the total system, which can be represented adequately only with a compliant system model.

Though not strictly a driveability phenomena, a representation of engine knock was made incorporating the trends due to spark advance, engine speed and load; and was characterised particularly by the bounds of the engine map where knock was an implicit problem.

Both the hesitation and the knock characteristics are evaluated by the function subprogram FNCDRV.

5.5 C.A.S.S. PROGRAM ELEMENTS.

5.5.1 Simulation Main Program (SIMUL).

Figure 5.3 illustrates the logical flow of the main program SIMUL; a listing of which is given in Figure 5.4.

At the beginning of each run the constants and system variables are initialised by the subroutines INICON and INIVAR. These constants are outputted to a disc file in ascii format by PRTCON. After the status flags have been initialised the simulation loop is entered.

At this stage the automatic driver is off and the system initialisation begins. Much of the major loop is bypassed (only subroutine SYSTEM being accessed) until the steady state criterion is met; this is indicated by a steady air-fuel ratio (line 95 in Figure 5.4). Subroutine INIGER then reinitialises certain variables and selects first gear. The automatic driver is then switched on and the simulation begins from point A (Figure 5.3).

Subroutine EPAVEL fetches the vehicle velocity that is desired and the automatic driver (subroutine AUTO) chooses a throttle angle predicted to achieve this within the time step of the program loop. AUTO must also manage the clutch, gear ratio and brakes. The nominal air-fuel ratio and ignition advance for the present engine operating point is defined within subroutine CNTRLS. The performance of the vehicle is now simulated from the given controls and present state by subroutine SYSTEM; updating the vehicle state (speeds, torques, etc.). Engine temperatures are not necessarily updated each time around the major loop, but at specified longer time intervals, because of their relatively slowly varying nature. Finally the distance travelled, cumulative fuel and emissions are updated using the vehicle speed, and engine flows.

The program then loops back to point A unless it is the end of the schedule or time to output the state vector. This latter is performed by subroutine OUTVAR which writes the variables to a binary file in a format suitable for interactive data analysis. If it is the end of the driving schedule then the subroutine OUTVAR prints the summary statistics before the program ends.

5.5.2 Initialisation of Constants (INICON).

Most constants associated with the simulation are contained in separate files as can be seen from the pseudocode of the subroutine INICON (Figure 5.5). Such changes as say, vehicle mass or driving schedule, can then be implemented easily without re-compilation of the program module.

Full use is made of the file handling routines contained in the PRIME applications library VAPPLB. Routines such as OPNP$A interrogate the user for the name of the file to be opened, and automatically flag the success or failure of the operation. If any of the files cannot be opened correctly or the data is corrupted then INICON sets a status flag and returns control to the main program SIMUL.

Input files are requested for the vehicle data, the driving schedule, and the control coefficients:

Vehicle Data File.

The first two lines of the file contain any comments about the simulation run that the user wishes to be entered in the output file. If the comments contain the word 'hot' then a status flag is set to indicate a 'hot-start' simulation is to be performed. The remainder of the file contains constants mainly associated with the engine, gearbox, vehicle and driver. In addition the file contains the communication interval for recording the program variables.

Driving Schedule File.

The first item of data is the number of cycle velocities contained in the file. These are specified at one second intervals for the schedule and are supplied and stored in integer form. The subroutine EPAVEL (see section 5.5.4) scales the numbers by 0.05 in order to obtain the desired vehicle velocity in ms^{-1}.

Control Coefficient File.

The file contains the regression coefficients for obtaining the engine controls (ignition advance and air-fuel ratio) as a function of the engine variables. Any constant perturbation of a control from that given by the regression must also be provided.

Several other constants are calculated by INICON. These are associated chiefly with the vehicle and with temperature integrations.

On returning from INICON all files are closed.

5.5.3 Initialisation of Variables (INIVAR).

The pseudocode for INIVAR (Figure 5.6) illustrates the main variable groups to be initialised prior to simulation.

The engine speed is set to 800 r.p.m. with the throttle at its idle position and the gearbox in neutral. Engine controls are set appropriately and the vehicle speed assumed zero. Engine temperatures are initialised to the ambient temperature unless a 'hot start' is indicated (see section 5.5.2); in which case they are given values which correspond approximately to the steady-state temperatures achieved at a constant 40 m.p.h. in top gear. The steady state temperatures are determined by reference to the regression models implicit in the function subprograms FNCINL, FNCCYL and FNCEX.

The user is prompted for a file name for output of the simulation summary. The name is stored for use by PRTCON (see section 5.5.4) and a file is created called 'name.HDR' containing header information for the 'data analysis' file.

The Warwick University package IDA (Interactive Data Analysis) requires data in two files of specific format: a binary file 'name.DAT' containing data, and an ascii file 'name.HDR' containing header information associated with the data file (Figure 5.7). On returning from the subroutine the IDA file is open to allow recording of the system variables by subroutine OUTVAR.

5.5.4 Driving Schedule Translation.

The simulation model is exercised by a driving schedule (see section 3.1.3) which is stored in an integer array by the initialisation subroutine INICON (section 5.5.2). The subroutine EPAVEL (Figure 5.8) is called during every loop of the main program SIMUL to unpack the appropriate velocity; the velocities being specified for each second of the schedule.

As the system time approaches that for which the current schedule velocity is specified then the next cycle velocity is unpacked and used instead. The point at which this occurs is determined by the automatic driver 'look-ahead' parameter. If this was not done a small error between the actual and desired velocities would require a very large steady-state torque to achieve the desired velocity at the end of the time step (see section 5.5.5).

The cycle velocity is provided by EPAVEL in the real variable VEPA and is used by the automatic driver subroutine AUTO. If this is the last point specified then the logical variable QCYEND is set to flag the condition to the main routine SIMUL.

5.5.5 The Automatic Driver Module.

The algorithm used in the driver module to select the transmission gear, clutch setting and throttle angle has been discussed in section 4.6

Many automotive simulations used in fuel economy studies assume that the desired velocity has been attained at the end of the one second time step (5.11, 5.12). CASS differs from these in that the system is integrated over small increments of time to simulate the actual velocity and acceleration. It thus requires a closed loop in order to follow the driving schedule and is similar in principle to the method used by Beachley and Frank (5.13, 5.14) for an automatic transmission car.

Subroutine AUTO.

Figure 5.9 is a simplified flow chart of the driver subroutine AUTO; the kernel of the subroutine is the gear selection algorithm, which attempts to select the highest gear within the engine speed range that is easily able to supply the necessary torque.

The desired acceleration $\dot{V}_d$ (see equations 4.22 and 4.23) is used to calculate the torque required at the wheels. If two successive schedule velocities are zero then $\dot{V}_d$ is chosen to cause the vehicle to declerate to standstill at the end of the current time period.

Having calculated the engine speed and required engine torque for each gear, as detailed in equations 4.25 to 4.27, the highest gear within the engine speed range is chosen as a trial gear. The values of the engine controls at wide-open throttle (WOT) are then determined from the calibration functions (see section 5.5.6); in other words, the air-fuel ratio and spark advance are obtained for the maximum torque condition at the engine speed applicable to the trial gear. The function subprogram AMAXTQ then gives the maximum available torque for that gear.

The choice of the 'best' gear follows closely that described in section 4.6.4; with the subroutine SLIP being called if first gear is chosen and either: the engine speed would be too low without clutch slip; or insufficient torque can be provided by the engine at the given speed. The minimum engine torque is given by the function subprogram AMINTQ: if this is insufficient to retard the vehicle (if this is required) then the braking force (equation 4.30) is calculated and the throttle is closed. If the engine is able to supply the required torque then the subroutine THROTL is called to calculate and set the required throttle angle, and the braking force is set to zero before returning from the

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When a gear change is to be performed the clutch is disengaged and the transmission set to neutral. A counter is set to simulate the time taken to select the new gear: this is decremented on entry to the AUTO routine; and the throttle angle is ramped to the idle setting at the start of this period. With the gear in neutral the force required to meet the vehicle acceleration requirements is calculated to give the braking force as in equation 4.34.

The automatic driver normally prevents a gear change taking place too soon after the previous change: this is facilitated by a counter and is overridden in cases of engine overspeed or underspeed. Generally a minimum of 4 seconds between gear changes was found to be suitable.

Subroutine SLIP.

The algorithm for selecting clutch speed ratio is described in section 4.6.4 and figure 4.10. No dynamics have been associated with clutch actuation and the subroutine is called by AUTO only if first gear is involved.

Maximum torque at any given engine speed is obtained from the function subprogram AMAXTQ, after functions FNCAFR and FNCSA have supplied the nominal air-fuel ratio and spark advance for that engine speed.

Subroutine THROTL.

This routine is called from AUTO which provides the engine speed, engine controls, and the desired engine torque. A suitable throttle angle is obtained from function FNCTHR (see below) and this is then limited according to the maximum permitted rate of change of throttle.

Function Subprogram FNCTHR.

FNCTHR returns a value of throttle angle which gives the required engine torque for given engine speed and engine controls. If the engine is unable to supply sufficient torque in a positive or negative sense, then the throttle angle is returned as its maximum or minimum value respectively (Figure 5.10).

Figure 5.11 gives a listing of the routine which determines the required throttle angle from the torque characteristic by the method of bisection. The torque characteristic itself is monotonic in throttle angle (Figure 4.11); increasing with increasing throttle. The stopping criteria was three-fold initially: based on the function values, the range of throttle angle and the number of iterations as seen in Figure 5.11; the first of these alone, however, was found to be adequate.

5.5.6 Engine Control Functions.

Air-fuel ratio (AF) and spark advance (SA) are specified as polynomial functions of certain engine variables. The coefficients are read from a file by the subroutine INICON (section 5.5.2) and held in a common block for use by the function subprograms FNCAFR and FNCSA. The listing of FNCSA is shown in figure 5.12.

For ignition advance, engine speed and throttle angle are received as arguments; but the control functions are compiled and loaded separately from other routines to enable additional engine variables to be used if necessary, by accessing them via the common block. The air-fuel ratio function also used engine speed and throttle angle, passed as arguments; while coolant temperature was accessed as a common variable.

A full second or third order polynomial expansion was available. The choice depends on the logical states of QSA3 and QAF3, which are specified with the coefficients during initialisation. For certain simulation runs it was necessary to perturb the controls; for this a constant offset can be specified at the same time.

The subroutine CNTRLS evaluates the manifold absolute pressure (MAP) prior to invoking FNCAFR and FNCSA; as often the control functions are specified with reference to M.A.P. This is owing to the direct (to a first approximation) relationship between MAP and engine load (figure 5.13); there being at present no simpler, cheaper, or more readily available means of sensing this variable in volume production vehicles. The function subprogram FNCMAP contains a regression polynomial expression for inlet manifold absolute pressure in terms of engine speed, ignition advance, air-fuel ratio and engine torque.

5.5.7 Simulation of Engine and Vehicle Dynamics.

During each loop of the main program SIMUL (section 5.5.1) the system dynamics, with the exception of the engine temperatures, are integrated.

The subroutine SYSTEM invokes other routines to facilitate the integration of the vehicle speed and the transient fuel flow, which use an integration step independent of the time step for the rest of the systems. Subroutine INTGRT evaluates the cumulative states such as distance travelled, mass emissions and fuel consumption, during each program loop. The system temperatures are integrated by routines WATER, INLET, CYLIND and EXHST for the coolant, inlet manifold, cylinder surface and exhaust manifold temperatures respectively. The coolant temperature was integrated at 10 second intervals, while 1 second was chosen as the step size for the latter three temperatures.

Vehicle Speed.

The road load force (defined as the force required at the tyre-road interface necessary to maintain a particular constant speed) is

$$F_r(t) = C_R Mg + \frac{1}{2} A_f \rho C_D V(t)^2 + F_B(t) \tag{5.2}$$

from equation 4.1. This includes terms for the rolling resistance, aerodynamic drag and the braking force specified by the driver.

The force exerted at the wheels by the engine, the tractive force, is

$$\begin{aligned} F_w(t) &= T_{co}(t).\frac{R(g_i).R_a}{r_w}.\eta(g_i)\eta_a \quad , V_d(t) \geq V(t) \\ &= T_{co}(t).\frac{R(g_i).R_a}{r_w}.\frac{1}{\eta(g_i)\eta_a} \quad , V_d(t) < V(t) \end{aligned} \tag{5.3}$$

where T_{co} = clutch ouput torque (Nm)

$R(g_i)$ = gearbox ratio in gear g_i

R_a = final drive ratio

r_w = tyre rolling radius (m)

$\eta(g\ )$ = gearbox efficiency in gear g_i

η_a = final drive efficiency

V_d = desired vehicle velocity (ms^{-1})

V = simulated vehicle velocity (ms^{-1})

The net tractive force $(F_w(t) - F_r(t))$ gives the vehicle acceleration

$$\dot{V}(t) = (F_w(t) - F_r(t))/M_j(g_i) \tag{5.4}$$

as in equation 4.8.

Vehicle velocity is obtained by integrating equation 5.4 using a second order algorithm implemented in the form

$$V(t_k) = V(t_{k-1}) + \frac{\dot{V}(t_k) + \dot{V}(t_{k-1})}{2}.\delta T \tag{5.5}$$

where $t_k = k\delta T$, $k \in [0, 1, 2, ...]$

δT = integration time step (0.01 seconds)

The engine speed is then

$$\omega_E(g_i, t) = V(t).\frac{R(g_i).R_a}{r_w} \tag{5.6}$$

The simulation of the fuelling system involves the simple dynamic model of figure 4.5, and is described in section 4.4.5. The mass air flow is predicted by the subprogram FNCAIR, which models the flow as a function of engine speed, throttle angle, ignition advance and air-fuel ratio: this is implemented in a similar manner to FNCSA (section 5.5.6). As the air circuit is assumed to have negligible delay, it is only necessary to predict the transient fuel flow in order to determine the transient air-fuel ratio of the mixture inducted by the engine.

From figure 4.5 it is clear that the fuel flow into the induction manifold is the sum of the 'nominal' flow

$$\dot{m}_n(t) = K_{f/a}.\dot{m}_a(t) \tag{5.7}$$

and the throttle pump flow $\dot{m}_p(t)$.

Where $\dot{m}_a$ is the air mass flow, and $K_{f/a}$ is the nominal fuel-air ratio. The pump flow $\dot{m}_p$ being modelled as in equation 4.20.

The fuel mass inducted is thus represented as

$$m_f(s) = \frac{m_n(s) + m_p(s)}{(1 + \tau_c s)(1 + \tau_f s)} \tag{5.8}$$

where τ_c and τ_f are the time constants associated with the control and inlet manifold respectively. In state variable form this becomes

$$\dot{x}(t) = Ax(t) + Bu(t) \tag{5.9}$$

where x is a 2-vector

$$x_1(t) = \dot{m}_f(t)$$
$$x_2(t) = \dot{x}_1(t)$$

and

$$A = \begin{bmatrix} 0 & 1 \\ \frac{-1}{\tau_c \tau_f} & -\frac{\tau_c + \tau_f}{\tau_c \tau_f} \end{bmatrix}, \qquad B = \begin{bmatrix} 0 \\ \frac{1}{\tau_c \tau_f} \end{bmatrix}$$

and

$$u(t) = \dot{m}_n(t) + \dot{m}_p(t)$$

The system of equation 5.8 is implemented in discrete time form

$$x(t_{k+1}) = A_1 x(t_k) + B_1 u_k$$

as in Appendix B. The controller time constant τ_c was fixed and the fuel 'lag' represented by equation 4.21. However, A and B are re-evaluated only if

$$\frac{\tau_f - \tau_o}{\tau_o} > \varepsilon_f$$

where τ_o is the value τ_f when A and B were calculated previously, and $\varepsilon_f = 0.3$. The integration step for this model was 0.01 s.

Exhaust Emissions.

The transient air-fuel ratio is used by the subroutine EMISS to predict steady-state and transient (with respect to non-steady-state engine temperatures) emission flows. The steady state flows are predicted by regression models, and adjusted according to the engine temperatures (section 4.4.4). Cumulative emissions are obtained by integrating the flows in a similar manner to the vehicle velocity (equation 5.5).

Engine Temperatures.

The steady state engine temperatures are readily modelled by multiple linear regression analysis of the mapping data. The steady state coolant temperature was defined, however, by the engine thermostat.

The rate at which coolant temperature approaches that fixed by the thermostat, depends in part on the load on the engine and the cooling effect due to vehicle speed. A second order model was chosen with time constants

$$\tau_1 = 100 \text{ s}$$

$$\tau_2 = 120 - 60.\dot{F}(t)/\dot{F}_{max} + V(t)/3 \quad \text{s}$$

where $\dot{F}_{max}$ is the maximum value of fuel flow

$V(t)$ is the vehicle velocity (ms^{-1})

The step length for integrating coolant temperature was 10 s: the state transition matrix of the discrete implementation was updated only when τ_2 differed from the time constant used to compute the existing matrix by at least 5 s.

Equilibrium cylinder surface temperature differs from the steady-state condition if the coolant temperature has not stabilised; that is if coolant temperature is 10°C below steady-state, then cylinder surface temperature at equilibrium will be about 10°C below its steady-state value (5.6, 5.7). A first order model with a time constant of 10 s was used to represent the dynamics, and the temperature was updated every 1 second.

Equilibrium inlet manifold surface temperature is dependent on the mixture flow in the manifold, and in this case the coolant temperature. At idling the temperature is virtually the same as the coolant but at full power it may be 40°C below this (5.10). Again a first order linear representation was used for the temperature dynamics; having a time constant of about 10 s and integration step length of 1 s. The equilibrium surface temperature for the induction manifold was of the form

$$T_i = T_{is}.(1 - C_i(T_{is} - T_c)/T_{cs})$$

where T_i = equilibrium inlet manifold surface temperature

T_{is} = steady state inlet manifold surface temperature

T_{cs} = steady state coolant temperature

T_c = coolant temperature

C_i = 0.75

5.5.8 Simulation Results.

C.A.S.S. provides output in the form of a binary record of certain system variables, and simulation run summaries in ASCII format.

Routine OUTVAR outputs a set of variables (table 5.2) in a binary form compatible with the Interactive Data Analysis (IDA) facility used at the University of Warwick. Output occurs at a communication interval specified by the user (see figures 5.3, 5.4), and represents the finest detail that can be observed subsequently. Two IDA files are necessary - a binary data file, and a header file containing variable names, units and basic statistics.

Certain model details and other information is written to an ASCII file by routine PRTCON (see figures 5.3, 5.4) and the run summary is appended to this by OUTPUT after the driving schedule has been 'driven'. The summary information, which includes cumulative fuel consumption and emissions, also appears on the user terminal. Figure 5.13 shows an example of the run summary output file.

TABLE 5.2

C.A.S.S. Output Variable List

Number	Description	Unit
1	Engine speed	r.p.m.
2	Engine net torque	Nm
3	Ignition timing (SA)	degrees btc
4	Air-fuel ratio (AF)	
5	Throttle angle	degrees
6	Manifold absolute pressure (MAP)	kPa
7	Mass air flow	kg/h
8	Mass fuel flow	kg/h
9	Carbon monoxide (CO) flow	g/h
10	Unburnt hydrocarbon (HC) flow	g/h
11	Nitrogen oxides (NOx) flow	g/h
12	Engine knock index	
13	Ingine hesitation index	
14	Coolant temperature	K
15	Inlet surface temperature	K
16	Cylinder surface temperature	K
17	Exhaust surface temperature	K
18	Vehicle speed	m/s
19	Vehicle acceleration	m/s**2
20	Brake setting	N
21	Gear number	
22	Clutch speed ratio	
23	Gear efficiency	
24	Nominal air-fuel ratio	
25	Driving schedule velocity	m/s
26	Road load	N
27	Vehicle inertial mass	kg
28	Road load (including braking)	N
29	Cumulative fuel consumption	g
30	Cumulative CO	g
31	Cumulative HC	g
32	Cumulative Nox	g
33	Distance travelled	m

5.6 OPERATING C.A.S.S.

The simulation program is implemented on a Prime 550 minicomputer running under the Primos operating system (5.15). The user has to supply the necessary data in free-format files and enter the file names as the program requests. The dialogue might be as follows (user input is underlined):

<u>RESUME *SIM</u>

File containing engine-vehicle data: <u>&DAT2H</u>

File containing driving schedule: <u>VELOC</u>

File containing control coefficients: <u>REFCNTRL</u>

Data file for output: <u>Z10</u>

... SIMULATION RUNNING ...

Usually, however, the simulation is performed in batch mode operation - when the user's input forms the job control file. The simulation output includes a run summary (figure 5.14) and an IDA data file containing the values of the variables logged at each communication interval. This data is then available for manipulation by special purpose programs, and for analysis using the Interactive Data Analysis (IDA) package.

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NOTATION

V_d desired vehicle velocity

V actual vehicle velocity

F_r vehicle road load force

C_R rolling resistance coefficient

M vehicle mass

g acceleration due to gravity

A_f vehicle frontal area

C_D vehicle aerodynamic coefficient

ρ air density

F_w tractive force

F_B braking force on vehicle

$M_j(g_i)$ inertial mass of vehicle in gear g_i

$R(g_i)$ ratio of gear g_i

R_a final drive ratio

r_w wheel rolling radius

$\eta(g_i)$ efficiency of gear g_i

η_a final drive efficiency

$X_1 \ldots X_k$ model regressors

$\hat{Y}$ estimate of Y

$b_0 \ldots b_k$ coefficients produced by regression

T_{co} clutch output torque

$\dot{m}_n$ nominal fuel mass flow

$K_{F/A}$ nominal fuel-air ratio

$\dot{m}_a$	air mass flow
$\dot{m}_f$	fuel mass flow
τ_c	controller time constant
τ_f	fuel circuit time constant
$x(t)$	state vector
$u(t)$	input vector
A	coefficient matrix
B	driving matrix
A_1	coefficient matrix (discrete time)
B_1	driving matrix (discrete time)
τ_o	value of τ_f corresponding to current A_1, B_1
τ_1	cooling system time constant
τ_2	cooling system lag
$\dot{F}_{max}$	maximum fuel flow
C_i	cooling system parameter

```
Regression of channel  4  (14 predictors)

-0.18107940D 00    0.15733212D 01   -0.92286246D-03    0.62963131D 00
 0.81577603D-02    0.64037689D-01    0.31524717D-05    0.99354255D-03
-0.46740376D-01   -0.75030302D-04    0.00000000D 00    0.00000000D 00
 0.00000000D 00   -0.16826788D-04    0.00000000D 00    0.00000000D 00
 0.00000000D 00   -0.23635083D-04    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00   -0.13494618D-06    0.27949522D-04
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00

Constant term = 0.48032289D 02

SSR= 0.23811466D 06   DFR= 0.14000000D 02   MSR= 0.17008190D 05
F  = 0.66089158D 04   SSD= 0.18529358D 04   DFD= 0.72000000D 03
MSD= 0.25735219D 01   SST= 0.23996759D 06   DFT= 0.73400000D 03

Standard error of estimate = 0.16042200D 01
Coeff of multiple corr.    = 0.99613171D 00
Coeff of multiple determ.  = 0.99227839D 00
Coeff of multiple determ
corrected for deg. of frdm.= 0.99212825D 00

T-statistic for coeffs:
-0.17090368D 02    0.28035130D 02   -0.80721830D 01    0.17656686D 03
 0.12052165D 02    0.35570556D 01    0.33437988D 02    0.11358312D 02
-0.10451508D 02   -0.54545915D 01    0.00000000D 00    0.00000000D 00
 0.00000000D 00   -0.48400427D 01    0.00000000D 00    0.00000000D 00
 0.00000000D 00   -0.15359314D 02    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00   -0.13999624D 02    0.44904990D 01
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
 0.00000000D 00    0.00000000D 00    0.00000000D 00    0.00000000D 00
```

Figure 5.1 Statistical output from regression modelling program for a restrained 3rd order MAP model

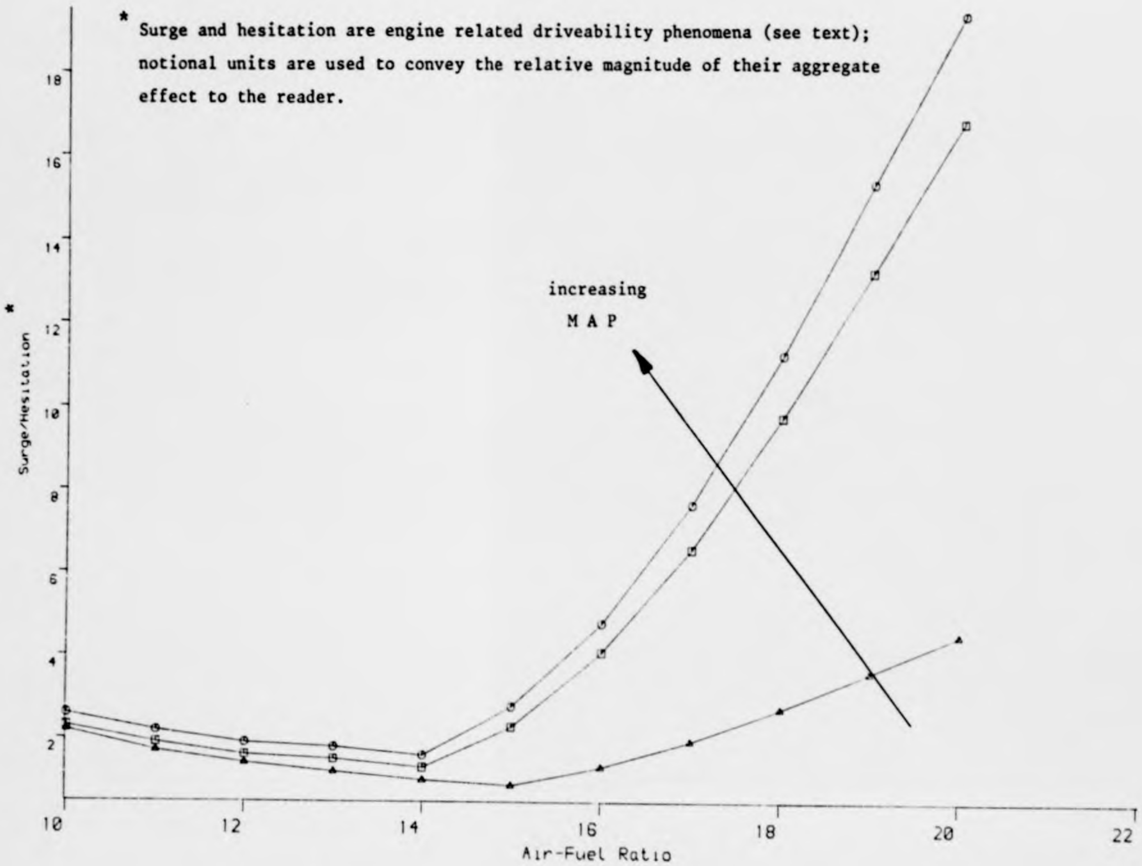

Figure 5.2(a) 'Driveability' (800 rpm)

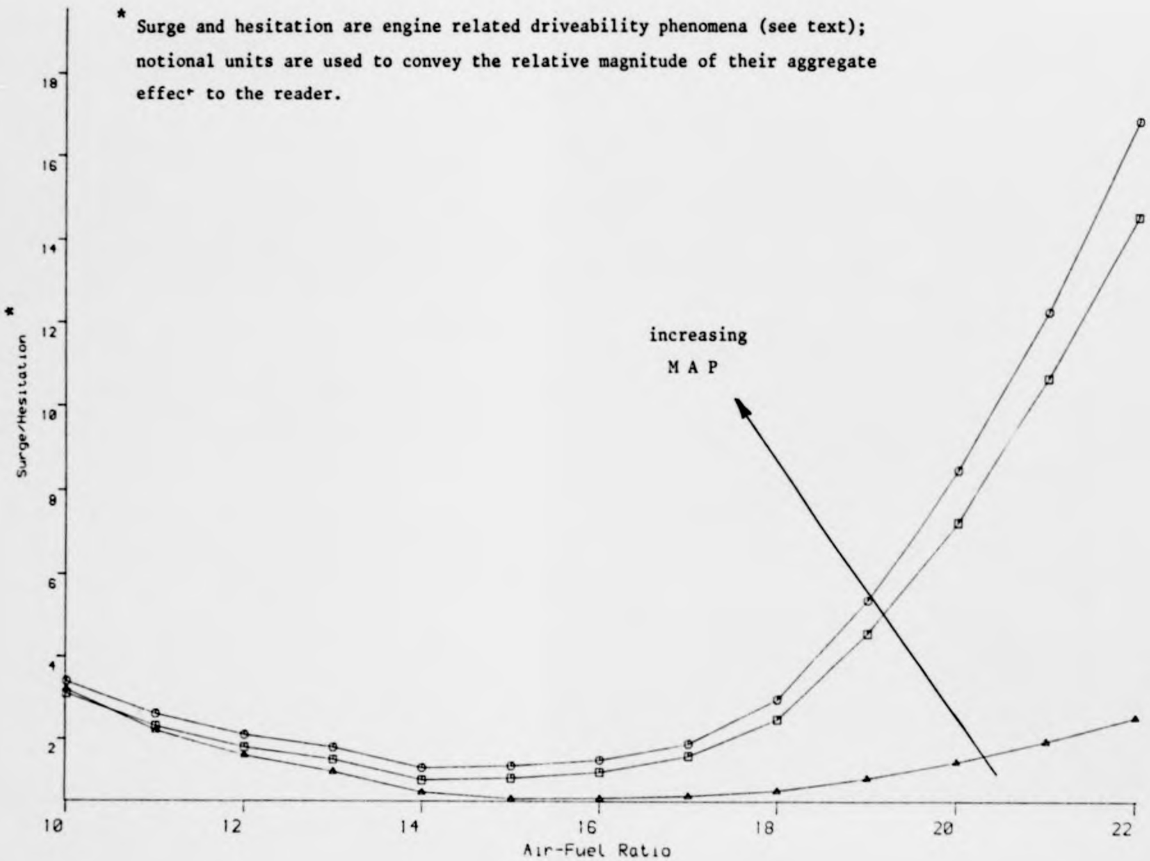

Figure 5.2(b) 'Driveability' (3200 rpm)

* Surge and hesitation are engine related driveability phenomena (see text); notional units are used to convey the relative magnitude of their aggregate effect to the reader.

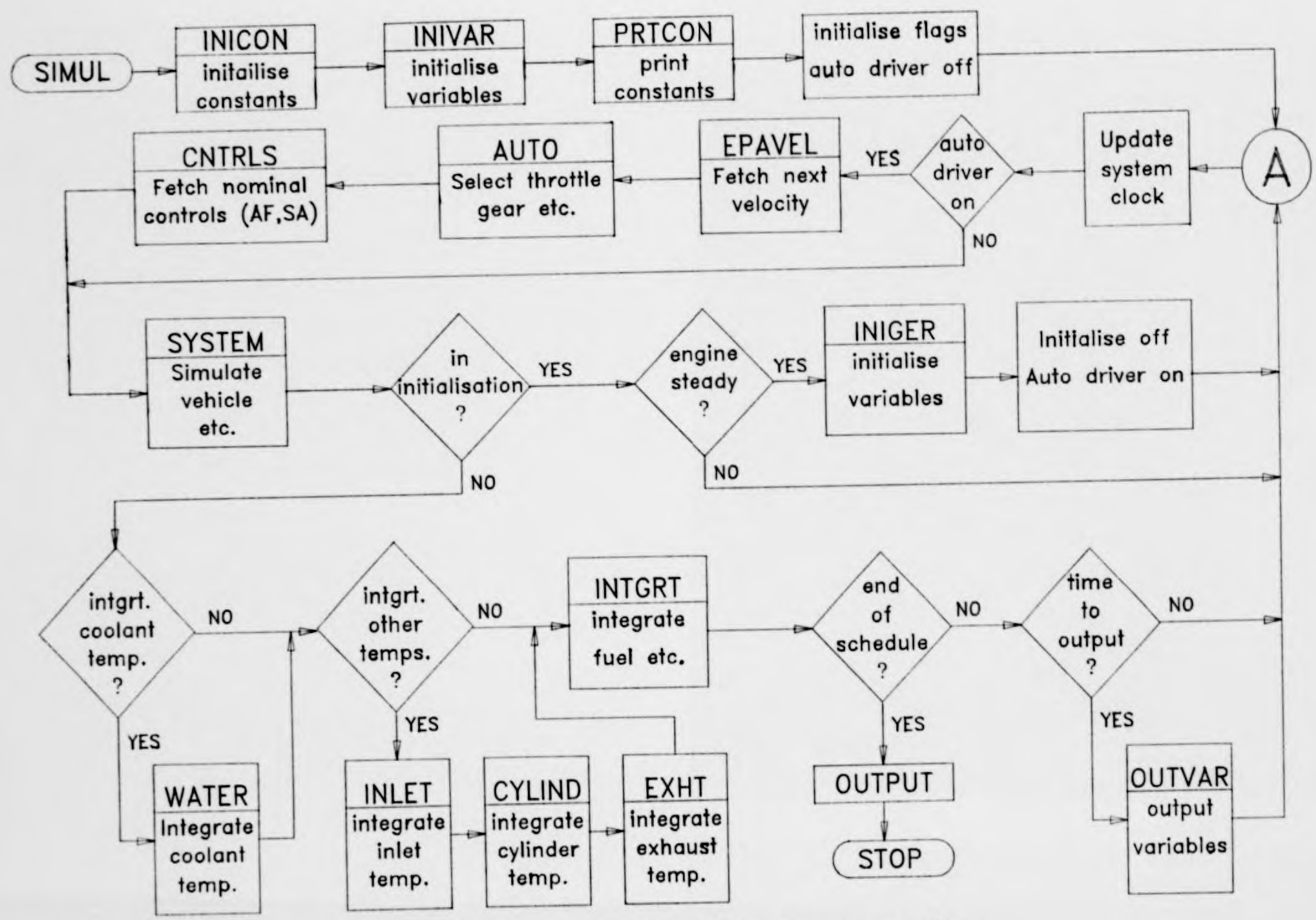

Figure 5.3 C.A.S.S. Simulation Main Program Flow

Figure 5.4(a) Simulation Main Program

```
C AUTOMOBILE SIMULATION MAIN PROGRAM



      SUBROUTINE SIMUL
$INSERT COMMON                        /* Load simulation common block

$INSERT SYSCOM>A$KEYS                 /* Load application library keys

C INITIALISE ALL CONSTANTS AND TABLES
   10 CALL INICON
      IF (Q) GO TO 15                 /* Check if initialisation satisfactory
      Q=YSNO$A(' ABORT PROGRAM',14,A$DYES)
      IF (.NOT.Q) GO TO 10
      PRINT 100
  100 FORMAT (//' **** ABORTED ****'//)
      GO TO 999



C BEGINNING OF RUN
   15 CONTINUE
C INITIALISE VARIABLES
      CALL INIVAR                     /* Initialise engine speed, etc.
      CALL PRTCON                     /* Print constants in output file
      QCYEND=.FALSE.                  /* Not end of driving schedule
      QINIT=.TRUE.                    /* Initialisation mode
      QAUTO=.FALSE.                   /* Auto driver off
C TIME STEP FOR SAVING ENGINE VARIABLES
      ISAVE=TSAVE/TIC+.5              /* No. of clock ticks between saving.
C INTERVAL BETWEEN COOLANT TEMPERATURE INTEGRATIONS.
      TCOOL=30.0                      /* Seconds
      ICOOLT=TCOOL/TIC+0.5            /* Equivalent clock ticks
      ICOOL=0
C INTERVAL BETWEEN INLT,CYLT,EXHT INTEGRATIONS
      TTEMP=1.0                       /* Seconds
      ITEMPT=TTEMP/TIC+0.5            /* Equivalent clock ticks.
      ITEMP=0


      PRINT 200
  200 FORMAT(//'   SIMULATION RUNNING ... '//)


```

Figure 5.4(b) Simulation Main Program

```
C BEGINNING OF MAJOR LOOP FOR EACH TIME STEP IN SCHEDULE.
   20 CONTINUE
C UPDATE SYSTEM CLOCK
      ZCLK=ZCLK+1
      TIME=ZCLK*TIC
      IF( NOT QAUTO) GO TO 30 /* Jump if auto driver off

C AUTOMATIC DRIVER
      CALL EPAVEL                /* Fetch next cycle velocity.
      CALL AUTO                  /* Subroutine to simulate driver
C   { AFNOM=f(RPM,TORQ,...) }
C   { SA   =f(RPM,TORQ,...) }
      CALL CNTRLS                /* Get nominal controls AFRNOM and SANOM
   30 CALL SYSTEM                /* Integrate system dynamics

C SKIP IF IN INITIALISATION
      IF(QINIT) GOTO 45          /* Are we in initialisation ?

C*********************************************/*
C                                            /*
C INTEGRATION SUBROUTINES                    /*
C Ready to integrate coolant ?               /*
      ICOOL=ICOOL+1                          /*
      IF(ICOOL LT ICOOLT) GO TO 42           /*
      CALL WATER                             /*
      ICOOL=0       /* Reset counter         /*
C Ready to integrate other temperatures ?    /*
   42 ITEMP=ITEMP+1                          /*
      IF(ITEMP LT ITEMPT) GO TO 44           /*
      CALL INLET                             /*
      CALL CYLIND                            /*
      CALL EXHST                             /*
      ITEMP=0       /* Reset counter         /*
   44 CONTINUE                               /*
C Integrate distance, emissions, etc         /*
      CALL INTGRT                            /*
C Check if end of schedule                   /*
      IF(QCYEND)GO TO 60                     /*
C Time to o/p state-variables ?              /*
      IF((ZCLK-ZCLKO) LT ISAVE) GO TO 20     /*
C OUTPUT SECTION                             /*
      CALL OUTVAR  /* Data file              /*
      ZCLKO=ZCLK   /* Time at last o/p       /*
      GO TO 20                               /*
C*********************************************/*
```

Figure 5.4(c) Simulation Main Program

```
   45 CONTINUE
C RESET INTEGRATION TIMER DURING INITIALISATION
      ZCLK1=ZCLK
C ENGINE IN STEADY STATE ? (AF stabilised ?)
      IF (ABS(AF-AFRNOM) GT. STEADY) GO TO 20
C Engine in steady-state ready for run
   50 CONTINUE
      QINIT= FALSE              /* Reset non-initialisation mode
      QAUTO= TRUE               /*Switch on auto driver
      CALL INIGER               /* Reset variables; set gear=1
      CALL OUTVAR               /* Output initial states
      GO TO 20
C ************ END OF RUN ******************
   60 CONTINUE
      CALL OUTVAR               /* Output state vector
      CALL OUTPUT               /* Output summary of run
C ***************** END ********************
  999 CALL DACLSE(101,IERR)    /* Close data analysis file
      RETURN
      END
```

Figure 5.4(c) Simulation Main Program

```
   45 CONTINUE
C RESET INTEGRATION TIMER DURING INITIALISATION
      ZCLK1=ZCLK
C ENGINE IN STEADY STATE ? (AF stabilised ?)
      IF (ABS(AF-AFRNOM) GT STEADY) GO TO 20


C Engine in steady-state ready for run
   50 CONTINUE
      QINIT= FALSE              /* Reset non-initialisation mode
      QAUTO= TRUE               /*Switch on auto driver
      CALL INIGER               /* Reset variables, set gear=1
      CALL OUTVAR               /* Output initial states
      GO TO 20




C *********** END OF RUN *****************
   60 CONTINUE
      CALL OUTVAR               /* Output state vector
      CALL OUTPUT               /* Output summary of run

C **************** END *******************
  999 CALL DACLSE(101,IERR)    /* Close data analysis file
      RETURN
      END
```

Figure 5.5 Pseudocode version of subroutine INICON

```
START
set time step for simulation main loop
prompt for file name and read vehicle parameters
IF error detected THEN
      print error message
      set error status
      return to main program
END IF
calculate and set additional system constants
set time constants for temperature and fueling system models
prompt for file name and read driving schedule data
IF error detected THEN
      print error message
      set error status
      return to main program
END IF
prompt for file name and read controller coefficients
IF error detected THEN
      print error message
      set error status
      return to main program
END IF
RETURN  (program constants set)
```

Figure 5.6 Pseudocode version of subroutine INIVAR

```
START

set initial conditions for idling

IF cold start required THEN

      set initial temperatures for a cold start

ELSE

      evaluate and set initial temperatures for a hot start

END IF

create Interactive Data Analysis (IDA) file for logging
variables, using file name supplied by user.

open IDA file for subsequent output

IF error detected THEN

      print error message

      flag error status

END IF

RETURN (program variables initialised)
```

Figure 5.7 Interactive Data Analysis Header file

	CHANNELS	SAMPLES	SIZE	FORMAT
HDR	34	6860	4	0
	NAME	UNITS	INCREMENT	START
1VAR	TIME	SECS	0.20000	0.00000
	NAME	UNITS	SCALE	
CH01	Engine spd	r.p.m.	1.0000	
CH02	Torque	Nm	1.0000	
CH03	SA	Degrees	1.0000	
CH04	AF	Ratio	1.0000	
CH05	Throttle	Degrees	1.0000	
CH06	MAP	kPa	1.0000	
CH07	Air flow	Kg/hr	1.0000	
CH08	Fuel flow	Kg/hr	1.0000	
CH09	CO flow	g/hr	1.0000	
CH10	HC flow	g/hr	1.0000	
CH11	NOX flow	g/hr	1.0000	
CH12	Knock	-	1.0000	
CH13	Hesitation	-	1.0000	
CH14	Coolant t.	K	1.0000	
CH15	Inlet t.	K	1.0000	
CH16	Cylinder t	K	1.0000	
CH17	Exhaust t.	K	1.0000	
CH18	Vehicle spd	m/s	1.0000	
CH19	Acceln	m/s**2	1.0000	
CH20	Brakes	N	1.0000	
CH21	Gear	-	1.0000	
CH22	Clutch slip	Ratio	1.0000	
CH23	Gear ratio	Ratio	1.0000	
CH24	Gear eff.	-	1.0000	
CH25	Nominal AF	Ratio	1.0000	
CH26	Schedule	m/s	1.0000	
CH27	FRICSM	N	1.0000	
CH28	Inert. Mass	Kg	1.0000	
CH29	ROADLD	N	1.0000	
CH30	Cum. Fuel	Kg	1.0000	
CH31	Cum CO	g	1.0000	
CH32	Cum. HC	g	1.0000	
CH33	Cum. NOx	g	1.0000	
CH34	Distance	m	1.0000	

Figure 5.8 Pseudocode version of subroutine EPAVEL

```
START

determine driving schedule data array index

delta-t = (time at which next schedule point specified - current time)

IF delta-t not greater than 'driver look-ahead' THEN
.
.     increment data array index
.
.     IF last point in driving schedule set flag
.
END IF

unpack data array element (schedule velocity)

interpolate to obtain 'desired' velocity at next simulation time step

RETURN  (driving schedule velocity determined)
```

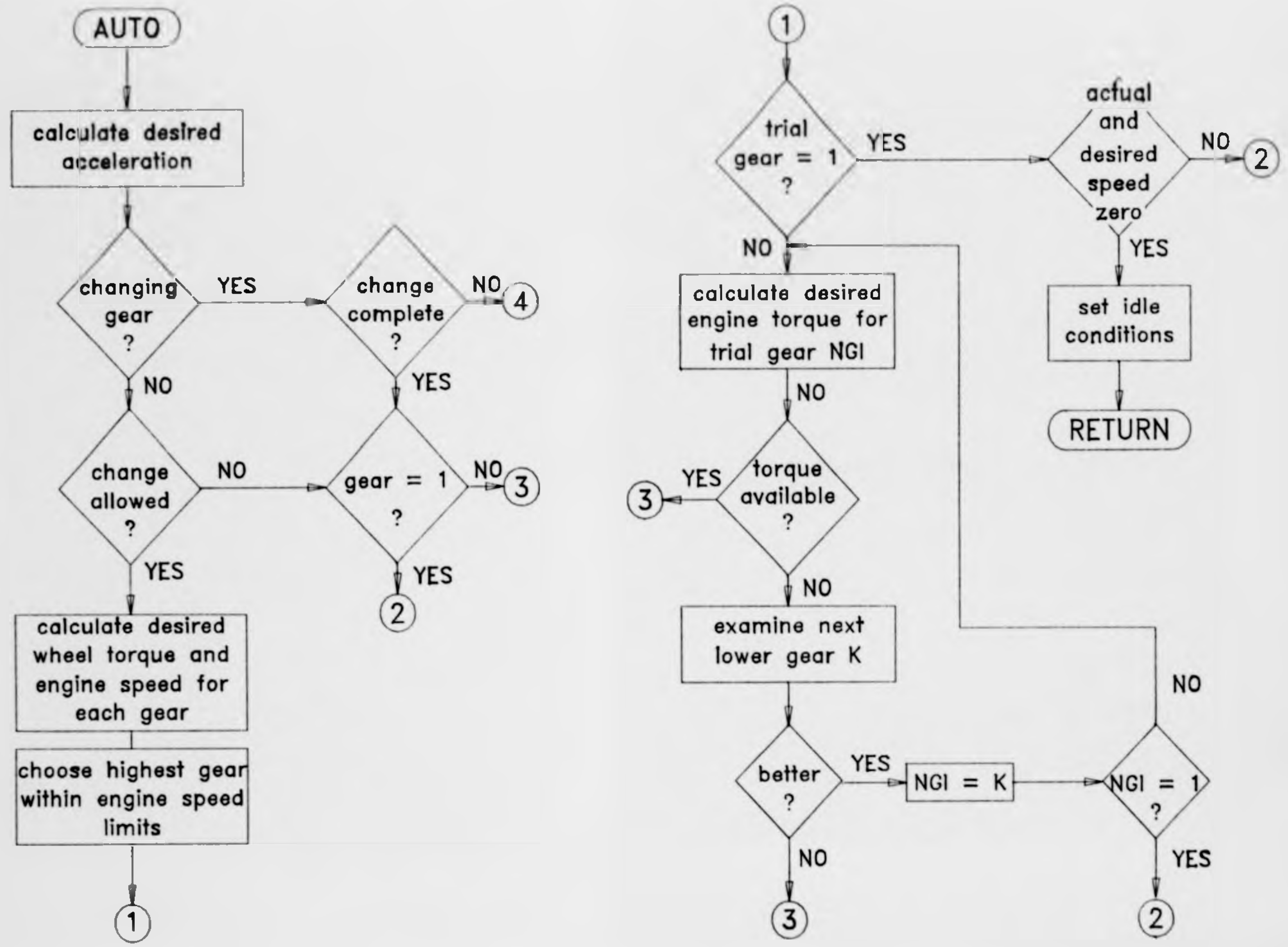

Figure 5.9a Flow chart of driver logic

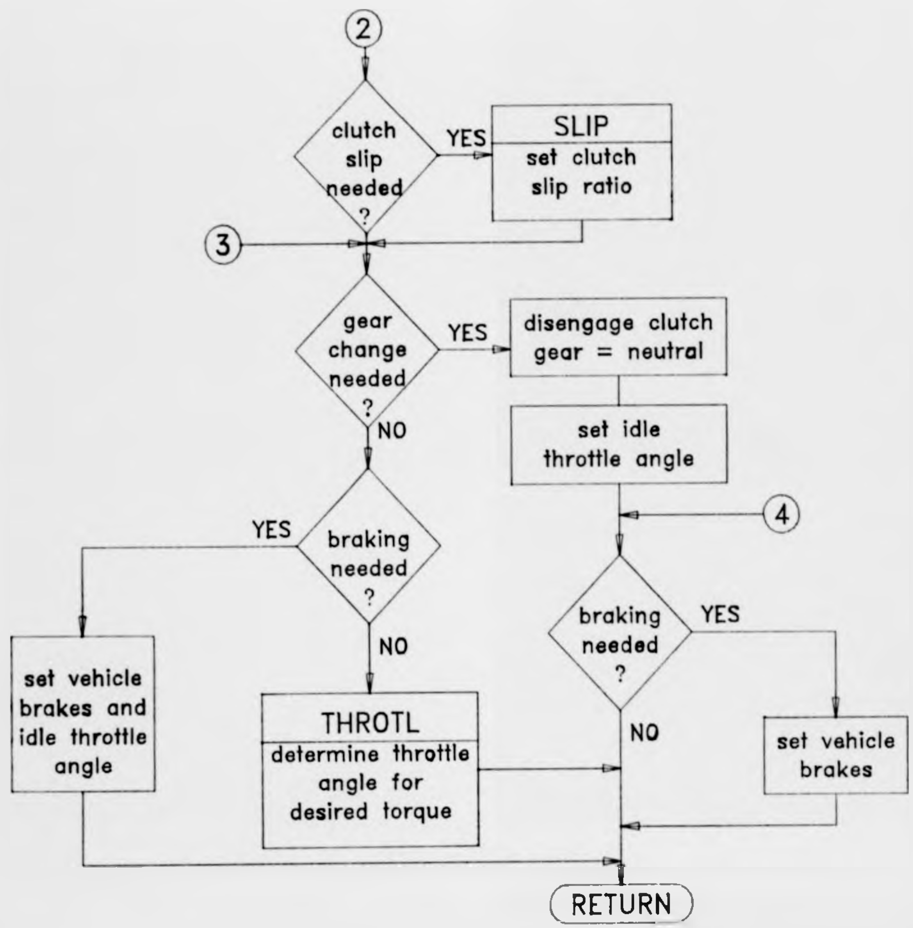

Figure 5.9b Flow chart of driver logic

Figure 5.10 Pseudocode version of subroutine FNCTHR

```
START

determine engine torque range at current engine speed

IF desired torque is in range THEN
.
.     DO WHILE convergence not achieved
.     .
.     .  choose throttle angle range to bracket the desired torque
.     .
.     .  bisect throttle angle range
.     .
.     .  determine torque at mid-point
.     .
.     .  IF range of throttle angle or torque very small on interval,
.     .  the procedure has converged
.     .
.     END DO
.
ELSE
.
.     IF desired torque less than minimum available THEN
.     .
.     .  set minimum throttle angle
.     .
.     ELSE
.     .
.     .  set maximum throttle angle
.     .
.     ENDIF
.
ENDIF

RETURN  ('desired' throttle angle determined)
```

Figure 5.11 Function to get throttle angle from FNCTQ by bisection

```
      FUNCTION FNCTHR(RPM,SA,AFR,TQ)

C     RPM    = engine speed
C     SA     = ignition advance (degrees b.t.d.c.)
C     AFR    = air-fuel ratio
C     X1     = lower bracket on throttle
C     X2     = upper bracket on throttle
C     FX1    = engine torque for throttle = X1
C     FX2    = engine torque for throttle = X2
C     XMID   = throttle angle at mid point of [X1,X2]
C     FMID   = engine torque at throttle = XMID
C     FNCTQ  = function returns engine torque
C     LOOP   = iteration count

      LOOP=0
      X1=5                                    /* Closed throttle
      X2=74                                   /* W.O.T.
      FX1=AMINTQ(RPM,SA,AFR)-TQ
      FX2=AMAXTQ(RPM,SA,AFR)-TQ
      IF (FX1*FX2.LE.0.0) GO TO 40            /* Check torque bracketing

C Not in range of torque
   10 IF(ABS(FX1)-ABS(FX2))20,20,30
   20 FNCTHR=X1
      RETURN
   30 FNCTHR=X2
      RETURN

C Torque bracketed by X1 and X2
   40 LOOP=LOOP+1
      XMID=(X1+X2)/2
      FMID=FNCTQ(RPM,SA,AFR,XMID)-TQ
      IF(FX1*FMID) 60,50,70

C Desired engine torque at XMID
   50 FNCTHR=XMID
      RETURN

C Desired engine torque bracketed by X1 and XMID
   60 X2=XMID
      FX2=FMID
      GO TO 80

C Desired engine torque bracketed by X2 and XMID
   70 X1=XMID
      FX1=FMID

C Test for convergence
   80 IF(ABS(FX1-FX2).GT.1.0) GO TO 40
C     IF(ABS(X1-X2).GT.0.5) GO TO 40
C     IF(LOOP.LT.5) GO TO 40

      FNCTHR=XMID
      RETURN
      END
```

Figure 5.12 Function specifying nominal SA as a control variable.

```
      REAL FUNCTION FNCSA(VR,VT)
$INSERT COMMON
      REAL*4 C(10)
      EQUIVALENCE (C(1),COESA(1))

C Coefficients are contained in C and the expansion in two variables
C may be 2nd. or 3rd. order. QSA3 = .TRUE. for 3rd. order.
C Any constant perturbation is contained in SAPERT.
C     VR     = engine speed
C     VT     = throttle angle
C     COESA  = array of regression coefficients for SA

      IF(QSA3) GO TO 10

C 2nd. order...
      FNCSA=SAPERT+C(1)+VR*(C(2)+C(6)*VT+C(4)*VR)+VT*(C(3)+C(5)*VT)
      RETURN

C 3rd. order...
   10 FNCSA=SAPERT+C(1)+VR*(C(2)+C(6)*VT+VR*(C(4)+C(9)*VT+C(7)*VR))
     &         +VT*(C(3)+VT*(C(5)+C(10)*VR)+C(8)*VT)
      RETURN
      END
```

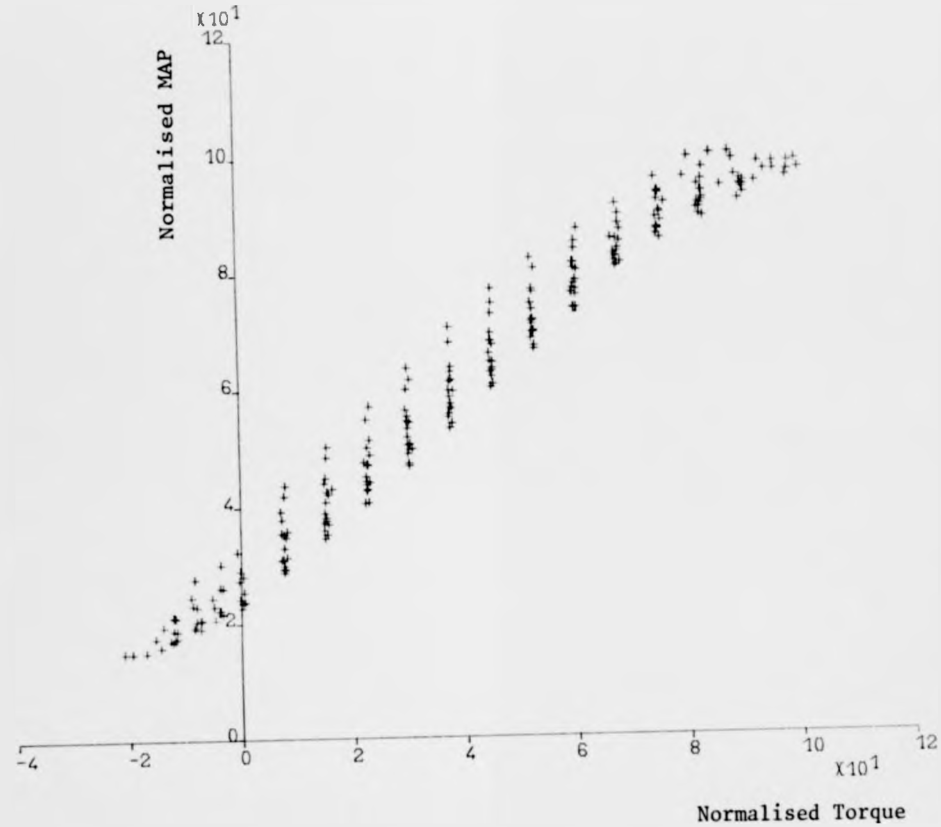

Figure 5.13 Relationship between steady state engine torque and M.A.P.

```
AUTO SIMULATION RUN
TUE, 22 JUN 1982
10 23 16

Reference run for HOT START  Vehicle data is from a Mark I 1300cc  Ford Escort.
Transmission ratios etc  from Vauxhall Astra 1600cc

                              ENGINE/VEHICLE PARAMETERS

--------VEHICLE CHARACTERISTICS--------- * -----------------------TRANSMISSION CHARACTERISTICS----------------------
                                         *
Vehicle Mass (Kg)                    987 * Gear                         NEUTRL    1      2      3      4      5      6
Frontal area (m**2)                1 755 * Gear ratio                          3 545  2 158  1 370  0 971
Aerodynamic coefficient            0 410 * Gear efficiency                      0 92   0 92   0 92   0 99
Coefficient of rolling friction    0 018 * Effective increase in
Tyre static radius (m)             0 266 * vehicle mass                  0 06   0 48   0 21   0 12   0 09
Brake constant                    1800 0 * Effective vehicle mass (Kg)   1046   1460   1194   1105   1075
                                         *
---------ENGINE CHARACTERISTICS--------- * Axle ratio                   3 740
Maximum engine speed (rpm)          5500 * Axle efficiency               0 92
Minimum engine speed (rpm)          1000 * Clutch torque transfer
Idle throttle angle (deg)              5 * ratio (disengaged)            0 50
Throttle pump                            *
    Rate (Kg/deg)                   0 04 *
    Limiting pump rate (deg )       50 0 * ---------AMBIENT CONDITIONS---------- * ----AUTO DRIVER CHARACTERISTICS-----
    Limiting throttle angle         65 0 *                                       *
Thermostat temperature (K)           358 * Air pressure (kPa)           101 35   * Driver look-ahead (sec.)        0 20
                                         * Air temperature (K)             293   * Gear shift duration (sec.)      1 60
                                         * Air density (Kg/m**3)         1 164   * Minimum time between shifts     4 00
                                         * Gradient                      0 000   *
                                         *                                       *

                                  SUMMARY OF RUN

File containing engine/vehicle data  &DAT2H
File containing driving schedule: VELOC
File containing control variable coeffs: E2CNTRL

Length of driving schedule: 1370 seconds.

Distance travelled (m) =         11833 588
Fuel consumed (Kg) =                0 795 (  1 111 litres)
Emissions produced (g):
               CO  =               65 018
               HC  =                8 140
               NOx =               16 336

Mean vehicle speed (m/s) =          8 638 ( 31 10 kph; 19 33 mph)
Mean fuel consumption (l/100km)     9 387 ( 30 10 mpg)

Mean exhaust emissions (g/km):
               CO  =                5 494
               HC  =                0 688
               NOx =                1 381
```

Figure 5 14 Example of C A B S Simulation Output

CHAPTER 6

STUDY OF AN OPTIMAL CONTROL APPROACH TO CONSTRAINED ENGINE OPTIMISATION

6.1 OPTIMAL CONTROL FORMULATION.

From the review of current approaches to the engine control problem (chapter2.2) it is clear that the techniques based on steady state engine maps, cannot address the effects due to temperature dynamics, actuator dynamics, cold-start phenomena, and catalyst light-off. An on-line approach reported by Dohner (6.1), however, attempted to account for some of these effects by using an engine and automatic gearbox mounted on a test bed; the driving schedule could then be 'driven' using a particular engine control law, and the resultant measurements used to compute an improved control by employment of an algorithm based on the formulation of a standard Bolza problem given by Bryson and Ho (6.2).

In the light of the potential advantages offered by such an on-line procedure with regard to system dynamics, it was decided to investigate some aspects of this type of calibration method. The formulation approach given below is more direct than that mentioned above, but yields a similar algorithm for obtaining a suitable engine calibration.

Formulation (first order gradient method).

The engine control problem is the common one of obtaining the engine controls (ignition advance SA and air-fuel ratio AF) as functions of some engine variables; such that fuel consumption is minimised over a specified driving schedule, and no more than a prescribed masses of gaseous pollutants are emitted. This type of problem was introduced in section 2.1

As the driving schedule is specified on the interval [0,T], the control u can be determined as a function of time; as can fuel and emission flows L and f respectively. Hence the problem may be written

Choose control

$$u = \begin{bmatrix} SA(t) \\ AF(t) \end{bmatrix} \tag{6.1}$$

to minimise the fuel consumed

$$J = \int_0^T L(u(t),t).dt \tag{6.2}$$

and meet the constraint on the cumulative emissions

$$e = \int_0^T f(u(t),t).dt \leq e^* \text{ , } e(0) = 0 \tag{6.3}$$

where e* represents the constrained values of hydrocarbons (HC), nitrogen oxides (NOx) and carbon monoxide (CO).

During a simulation the constraints (introduced in section 2.1) of the driving schedule, engine description and engine-vehicle dynamics will be met implicitly as the driving schedule is tracked.

From equation 6.2 we have a variation in the cost function (J)

$$\delta J = \int_0^T [\partial L/\partial u][\delta u(t)].dt \qquad (6.4)$$

due to an arbitrary change in the control $\delta u(t)$. The corresponding change in the terminal condition is

$$\delta e = \int_0^T [\partial f/\partial u][\delta u(t)].dt \qquad (6.5)$$

As 6.4 and 6.5 are linearised δJ has no minimum subject to constraints on the size of δe. In order to achieve this and limit the step size $\delta u(t)$, a quadratic integral penalty function in u(t) is applied and the problem addressed by adjoining the constraints with a vector of undetermined multipliers v.

$$\delta J = \int_0^T [L_u.\delta u + \delta u^T.W.\delta u + v^T f_u.\delta u].dt - v^T \delta e \qquad (6.6)$$

where W is an arbitrary 2 x 2 positive definite weighting matrix; $L_u = \partial L/\partial u$; $f_u = \partial f/\partial u$, and δe contains active constraints only.

The first variation of equation 6.6 gives

$$\delta(\delta J) = \int_0^T [L_u + \delta u^T W + v^T f_u]\delta(\delta u).dt \qquad (6.7)$$

which indicates that δJ has a minimum when

$$\delta u(t) = - W^{-1} [L_u + v^T f_u] \qquad (6.8)$$

Substituting 6.8 into equation 6.5 gives the predicted change in cumulative emissions

$$\delta e = -I_{eL} - I_{ee} v \qquad (6.9)$$

where

$$I_{eL} = I_{Le}^{T} = \int_0^T [f_u W^{-1} L_u^T].dt$$

and

$$I_{ee} = \int_0^T [f_u W^{-1} f_u^T].dt$$

The vector of Lagrange multipliers v is thus

$$v = -[I_{ee}]^{-1}[\delta e + I_{eL}] \qquad (6.10)$$

The predicted change in J found by substituting 6.8 and 6.10 into 6.4 is

$$\delta J = -(I_{LL} - I_{Le} I_{ee}^{-1} I_{eL}) + I_{Le} I_{ee}^{-1} \delta e \qquad (6.11)$$

where

$$I_{LL} = \int_0^T [L_u W^{-1} L_u^T].dt$$

Based on a reference trajectory $u^*(t)$ the algorithm described below interprets δe (6.9) as the reduction in e necessary to meet the terminal constraints. This is then used to evaluate the Lagrange multipliers (6.10) and a suitable change in the control histories (6.8) which will provide a reduction in cost or better satisfy the constraints.

As the solution is approached $\delta e \rightarrow 0$ and it is clear from equations 6.8, 6.10 and 6.11

$$\frac{\partial H}{\partial u} \rightarrow 0$$

$$v \rightarrow -I_{ee}^{-1} I_{eL}$$

$$I_{LL} - I_{Le} I_{ee}^{-1} I_{eL} \rightarrow 0$$

for

$$t \in [0,T]$$

where

$$H \triangleq L(u(t),t) + v^T f(u(t),t)$$

The Algorithm

a) Estimate the control u as a function of chosen engine variables.

b) Simulate a drive according to the specified velocity schedule using u as defined in (a). Record u(t), L(t), f(t) and the other engine variables.

c) Repeat step (b) twice perturbing a different element u_i during each replicate run. Record the same variables as in (b).

d) Compute the gradients L_u and f_u by finite differences using the reference and two perturbation runs.

e) Integrate the coefficients of equation 6.10 using the gradients found in step (d).

f) Calculate the vector of Lagrange multipliers from equation 6.10 where

$$\delta e = e^* - \int_0^T f(u(t),t)$$

g) Evaluate equation 6.8 to give the 'improved' control

$$u^{(i+1)}(t) = u^{(i)}(t) + \delta u^{(i)}(t)$$

h) Parameterise u(t) as a function of the chosen engine variables and repeat steps b to h until δe and δJ are sufficiently close to zero.

It should be noted that if equation 6.10 in step (f) produces $v_i < 0$, then that particular constraint is inactive. The corresponding row and column in the integrals of equation 6.10 should be removed, and the remaining multipliers re-evaluated.

6.2 PRACTICAL ASPECTS OF THE METHOD.

6.2.1 Basic Philosophy of Iterative Methods.

The basic philosophy of most numerical methods of parameter or trajectory optimisation is to produce a sequence of improved approximations to the optimum according to the scheme

i) Select an initial parameter vector X_i, $i = 0$

ii) Choose a suitable direction S_i which points in the general direction of the optimum.

iii) Choose a step length λ_i for movement in the direction S_i.

iv) Obtain the new approximation as

$$X_{i+1} = X_i + \lambda_i S_i \qquad (6.12)$$

v) Test X_{i+1} for optimality. If X_{i+1} is not the optimum, then increment i and repeat from step (ii).

This type of procedure is valid for constrained as well as unconstrained problems. The effectiveness of such methods depends on the efficiency with which the direction S_i and the step size λ_i are determined; whether the problem is of parameter optimisation, as illustrated for simplicity above, or of trajectory optimisation, as for the engine control approach formulated in section 6.1.

In the formulation of an algorithm intended for on-line determination of an engine control strategy it is essential to consider the high cost of obtaining the state trajectories of the system. This indicates a requirement for algorithms which converge in few iterations and require no more than first order gradient information - computational and data storage overheads are relatively unimportant. In addition it is desirable to avoid any need to supply a feasible first estimate of the engine control, as this may be difficult. The first-order gradient algorithm of section 6.1 satisfies this last criterion, but its rate of convergence is likely to depend considerably on the choice of step size for each iteration; as the step is taken in a direction that is downhill with respect to the Hamiltonian H.

6.2.2 Choice of the Weighting Matrix.

The quadratic integral penalty imposed in equation 6.6 results in the weighting matrix W^{-1} in equation 6.8, constraining the iteration step size.

Expanding

$$J = \int_0^T H(u(t),t).dt$$

to second order, about a particular trajectory we have

$$\delta J = \int_0^T \left[\frac{\partial H}{\partial u}(\delta u) + \frac{1}{2}(\delta u)^T \frac{\partial^2 H}{\partial u^2}(\delta u) \right].dt$$

$$\frac{\partial(\delta J)}{\partial(\delta u)} = \int_0^T \left[\frac{\partial H}{\partial u} + (\delta u)^T \frac{\partial^2 H}{\partial u^2} \right].dt$$

from which it is clear that δJ has a minimum when

$$\delta u(t) = -H_{uu}^{-1} H_u^T$$

where

$$H_{uu} = \begin{bmatrix} \frac{\partial^2 H}{\partial u_1^2} & \frac{\partial^2 H}{\partial u_1 \partial u_2} \\ \frac{\partial^2 H}{\partial u_2 \partial u_1} & \frac{\partial^2 H}{\partial u_2^2} \end{bmatrix}$$

positive definite. Hence a suitable choice for W is an approximation to H_{uu}, resulting in a crude quasi-Newton algorithm.

However in order to calculate H_{uu}, or indeed H_u it is necessary to have knowledge of the multipliers v (equation 6.10), which in turn cannot be calculated until W is determined. As the weighting matrix is an arbitrary positive definite matrix, and in order to avoid having to derive it iteratively, it is reasonable to use L_u in its determination.

Defining

$$W^{-1} = \alpha K \quad , \alpha \text{ scalar}$$

and

$$K = \begin{bmatrix} \left[\frac{\partial^2 L}{\partial (AF)^2}\right] \left[\frac{\partial^2 L}{\partial (SA)^2}\right]^{-1} & 0 \\ 0 & 1 \end{bmatrix}$$

which corresponds to the mean value of L_{uu}^{-1} on [0,T], if the off diagonal elements of L_{uu} are ignored, and the matrix is normalised with respect to the air-fuel ratio response. This K provides scaling to the controls and the step size α is chosen by some linear search technique.

Bryson and Ho (p225 ref.6.2) suggest that once a satisfactory weighting matrix such as

$$W = \varepsilon[\partial^2 H/\partial u^2] \quad , 0 < \varepsilon < 1$$

is obtained, this may be retained through subsequent iterations.

6.2.3 Computation.

Figure 6.1 shows the controls and driving schedule for the first portion of a reference run using the 'LA-4' schedule (section 3.1.3). The controls are parameterised in terms of certain engine variables using NAG routines to perform multiple linear regression analysis; in this case it can be seen that the engine dynamics cause air-fuel ratio excursions about a nominal (constant) ratio of 15.0.

The gradients in step (d) of the algorithm (see section 6.1) are evaluated using single sided finite differences. Where there is an engine speed or torque difference greater than 5% between reference and perturbation runs, the corresponding gradient is set to zero; this effectively eliminates its contribution to the computation of the new control. Screening such as this is important in removing the gross errors which may be perpetrated as a result of attempting to evaluate the gradients at a point where a gear change is occurring, for instance; at such points the change will inevitably occur at slightly different times for the reference and perturbation runs, owing to the difference between controllers. Additionally, if there is only a very small change in air-fuel ratio or ignition advance between reference and perturbation runs at a particular instant, then the gradient is set to zero at that instant to avoid prediction of extremely large and unlikely gradients.

In the calculation of the multipliers (step e of the algorithm), the two integrals I_{ee} and I_{eL} (equation 6.10) were evaluated using NAG routine D01GAF; and the multipliers determined using NAG routine F04ASF.

If an element v_i is negative then the matrices are reformed by deleting the ith row and column from I_{ee}, and the ith element from $(\delta e + I_{e\ell})$, before re-evaluating v. It is apparent from equation 6.10 that $I_{e\ell}$ must be of similar magnitude to δe, which means that W must be chosen appropriately. It is possible that there is a reduction necessary in one (or more) of the emissions ($\delta e_i < 0$), but the corresponding multiplier $v_i = 0$ - this implies that the new control will satisfy the constraint on that particular emission.

A desired control variation is calculated from equation 6.8 and added to the control trajectory taken from the reference run. This new control u(t) is then parameterised in terms of certain engine variables; the model being developed using multiple linear regression techniques involving NAG routines. The process begins with a full 2nd or 3rd order polynomial representation, which is progressively refined by eliminating the least-significant coefficients. Interaction by the engineer in the semi-automated process is essential in order to arrive at a suitable model. Modelling inaccuracies at this stage will inevitably result in deviations from the predicted performance of the new control, as will be assumptions about linearity of the system; this of course necessitates the iterative approach to a solution.

6.2.4 Results.

A number of simulations were performed to gain insight into the practicalities and performance of the optimal control algorithm described above. These tests included: weighting matrix determination; the effects of applying constraints to the permitted control adjustments; the effects of adjustments to the regression and gradient evaluation procedures; and the effect of different emission constraints.

Weighting Matrix

A matrix K (section 6.2.2) was determined using single-sided finite differences on a set of trajectories obtained by perturbing the engine controls either side of a reference control. Perturbations were of magnitude 5°c.a. ignition advance and 1.0 in air-fuel ratio for obtaining the gradients for most of the tests; these perturbations being known to give reasonable results on real engines. The resultant matrix

$$K = \begin{bmatrix} 16.01 & 0 \\ 0 & 1 \end{bmatrix}$$

In order to determine a reasonable scalar α for the weighting matrix crude linear searches were performed using a limited number of different emission constraints. This involved a reference and perturbation runs being performed, and new control laws being evaluated for a number of different values of α. Replicate simulation runs were performed using these new controls, and the results compared on the basis of fuel economy and emission performance.

Figure 6.2 summarises the reference and perturbation runs used and Figure 6.3 shows the normalised results of one typical study, where a constraint of

$$\begin{aligned} CO &\leqslant 6.07 \text{ g/km} \quad (9.97 \text{ g/mile}) \\ HC &\leqslant 0.731 \text{ g/km} \quad (1.18 \text{ g/mile}) \\ NOx &\leqslant 1.243 \text{ g/km} \quad (2.00 \text{ g/mile}) \end{aligned} \tag{6.13}$$

was used (i.e. an active constraint on NOx only). These results (Figure 6.3 and Table 6.1) and similar tests, led to a choice of $\alpha = 4$; giving, in this instance, satisfaction of the emission constraints in one iteration for no degradation of fuel economy.

TABLE 6.1

Results for Iteration Step Size Test.

Step Size α	Mean Fuel Consumption (l/100 km)	Mean Exhaust Emissions CO (g/km)	HC (g/km)	NOx (g/km)
0 (ref.)	9.346	6.028	0.689	1.714
1	9.380	5.172	0.630	1.218
2	9.370	5.030	0.623	1.223
3	9.349	4.903	0.618	1.230
4	9.333	4.786	0.612	1.238
5	9.341	4.740	0.609	1.248
6	9.323	4.653	0.606	1.253
7	9.303	4.568	0.601	1.260
8	9.283	4.482	0.598	1.264
9	9.263	4.412	0.595	1.265
10	9.248	4.351	0.592	1.268

Performance

In order to obtain an assessment of the performance of the algorithm on the engine control problem, several sets of simulation runs were performed using different emission constraints.

Figure 6.4 illustrates the results for 5 iterations with no effective constraint on the emissions. There is an almost linear reduction in fuel consumption and corresponding increase of nitrogen oxide emissions as the air-fuel ratio control tends towards leaner operation. The improvement of only about 7% in economy for rather large (about 30%) increases in nitrogen oxides, indicates that for this simulated automotive system, it may be difficult to find a control which accommodates a NOx constraint (below that of the reference run) and gives a worthwhile economy improvement.

Applying the constraint of equation 6.13 gave the results depicted in Figure 6.5 and Table 6.2. Here the constraint is met on the first iteration (by a reduction of 28% in NOx) for no change in fuel consumption. However, for subsequent iterations the procedure allows the nitrogen oxides to creep above the constraint boundary, while minimally reducing the fuel consumption. The hydrocarbon and carbon monoxide emissions remained below their respective constraint levels throughout.

TABLE 6.2

Results for NOx Constrained Problem.

Iteration Number	Mean Fuel Consumption (l/100 km)	Mean Exhaust Emissions CO (g/km)	HC (g/km)	NOx (g/km)
0 (ref.)	9.346	6.028	0.689	1.714
1	9.333	4.786	0.612	1.238
2	9.267	4.333	0.599	1.257
3	9.226	4.018	0.588	1.263
4	9.221	3.901	0.596	1.301
5	9.174	3.683	0.585	1.299

The fuel and emissions performance of any vehicle is bounded by physical limitations imposed by the design and condition of the hardware, in particular the engine and exhaust system. Clearly it is possible to define an emission constraint for a particular vehicle which cannot be satisfied by its current hardware. A number of simulations were performed using more stringent emissions constraints to ascertain the behaviour of the optimal control algorithm. Typical results for one such test are given in Figure 6.6 and Table 6.3. For this exercise the same reference run was used and a constraint of

$$\begin{aligned} CO &\leqslant 5.590 \text{ g/km} \quad (9.0 \text{ g/mile}) \\ HC &\leqslant 0.186 \text{ g/km} \quad (0.3 \text{ g/mile}) \\ NOx &\leqslant 1.243 \text{ g/km} \quad (2.0 \text{ g/mile}) \end{aligned} \tag{6.14}$$

was imposed.

In this exercise it is the hydrocarbon emissions which cannot meet the constraint, while the carbon monoxide and nitrogen oxides easily comply. In an attempt to determine a feasible control the fuel consumption is forced to increase dramatically; the effect on the hydrocarbons is not the expected corresponding monotonic decrease in hydrocarbons. In fact after the large improvement achieved by the first iteration, subsequent iterations degrade both fuel economy and hydrocarbon emissions, revealing poor convergence characteristics under these conditions. The procedure terminated at the third iteration as the computations failed to provide a significant control variation.

TABLE 6.3

Results for Problem with Severe HC Constraint

Iteration Number	Mean Fuel Consumption (l/100 km)	Mean Exhaust Emissions CO (g/km)	HC (g/km)	NOx (g/km)
0 (ref.)	9.346	6.028	0.689	1.714
1	12.315	2.712	0.392	0.472
2	14.026	2.391	0.477	0.470
3	15.166	2.040	0.456	0.550

Effect of Perturbation Size on Performance

The summary statistics given for the reference and perturbation runs (Figure 6.1) show some very large differences in cumulative emissions. This may indicate that the chosen perturbations (1.0 AF, 5° c.a.) are simply too large, even though they are known to be reasonable for similar work on actual engines, where measurement noise may be relatively high.

The repetition of several exercises with small perturbations (0.5 AF, 1° c.a.) used to obtain gradients, did not improve the performance of the procedure; in fact in most cases a degradation was noted. The reason for the poorer performance can be attributed to the fact that the considerable difference in the cumulative emissions is due, not to large perturbations, but to the disproportionate effect of relatively short times spent in engine operating conditions of high emission flow. For this reason the former choice for air-fuel ratio and ignition advance perturbations was used for most of the investigations.

Effect of Limiting Control Adjustments

In the optimal control approach used by Dohner (6.1) the control adjustment at each iteration was screened to avoid undesirable influence by suspect data. This screening took the form of ignoring any indicated control adjustment larger in magnitude, than three times the sample standard deviation of the adjustment over the entire test.

The comparative effect on the optimisation procedure of imposing such a constraint was investigated here using the simulation model. A number of runs were performed to assess the effect of varying the constraint on the control adjustment. Using the emission constraint of equation 6.13, typical results are shown in Table 6.4 for the first iteration.

TABLE 6.4

Typical results illustrating the effect of adopting a constraint on control adjustment.

Simulation Run	Fuel Flow (g/km)	Emission Flow CO (g/km)	HC (g/km)	NOx (g/km)	Control Adjustment Limit
A. Reference	9.35	6.03	0.689	1.71	-
B. 1st iteration	9.32	5.08	0.616	1.41	3 standard deviations
C. 1st iteration	9.32	4.99	0.612	1.42	7 standard deviations
D. 1st iteration	9.33	4.79	0.612	1.24	none

In this example, the screening of the control adjustments by this method, results in much poorer performance. Without the control limit (run D) the NOx constraint (which was violated by the reference run A) is met in a single iteration. This result was typical for the first iteration using different emission constraints, but is not as noticeable in subsequent iterations. The large effect noted can be attributed to the fact that the relatively few points along the driving schedule, discarded by the application of the constraint, have a major contribution to achieving the predicted reduction in emissions.

The need for such a constraint must be doubted on a number of counts:

a) The control adjustment for the iteration is calculated taking into account the predicted gains afforded by the (subsequently) discarded adjustments.

b) Certain doubtful gradients have already been eliminated from the procedure if engine speed or torque differs by more than 5% from reference to perturbation runs.

c) The regression procedure has a smoothing effect which would tend to reduce the magnitude of any outlying indicated control adjustments.

Exclusion of Data from the Regression

The regression of the new (time-varying) control in terms of engine variables is a semi-automated procedure, not an automatic one (Section 6.2.3); the human brain is remarkably good at comparing data such as the indicated control adjustments and a corresponding regression model, when they are manipulated and displayed in graphical form by the computer.

During the above optimal control procedure, a number of indicated control adjustments will be zero due to the discarding of certain gradients where engine speed or torque differs by greater than 5% between reference and perturbation runs; such as may occur in the proximity of a gear change. It is important that these zero points are eliminated from the regression to avoid biasing the model, and thus degrading the control. Table 6.5 illustrates such an effect: the exercise is the same as for Table 6.4; run A is the reference, run B is the first iteration where the regression model does not include the discarded gradients (as run D in Table 6.4), run C is based on a model which includes control adjustments that have been 'set' to zero.

TABLE 6.5

Results illustrating the effect of bias in the engine control model.

Simulation Run	Fuel Flow (g/km)	Emission Flow CO (g/km)	HC (g/km)	NOx (g/km)	Comment
A. Reference	9.35	6.03	0.689	1.71	-
B. 1st iteration	9.33	4.79	0.612	1.24	'correct' model
C. 1st iteration	9.37	5.24	0.667	1.47	biased model

These results clearly illustrate the magnitude of the effect of the bias, due to incorporation into the control regression model, of data points which were not involved in the calculation of the indicated control adjustments. The effect is in fact comparable, to that (illustrated in Table 6.4) of ignoring indicated control adjustments corresponding to gradients which have entered into the calculation of the improved control.

REFERENCES

6.1 Dohner, A.R., 'Transient system optimisation of an experimental engine control system over the Federal emissions driving schedule', SAE Paper 780286, 1978.

6.2 Bryson, A.E. and Ho, Y.C., 'Applied optimal control', Book Publ. Halstead Press, 1975.

NOTATION

$u(t)$	input vector
J	cost function
L	fuel flow
e	vector of cumulative emissions
e^*	vector of emissions constraints
f	vector of emission flows
W	weighting matrix
v	vector of Lagrange multipliers
H	Hamiltonian
K	weighting matrix
α	step size

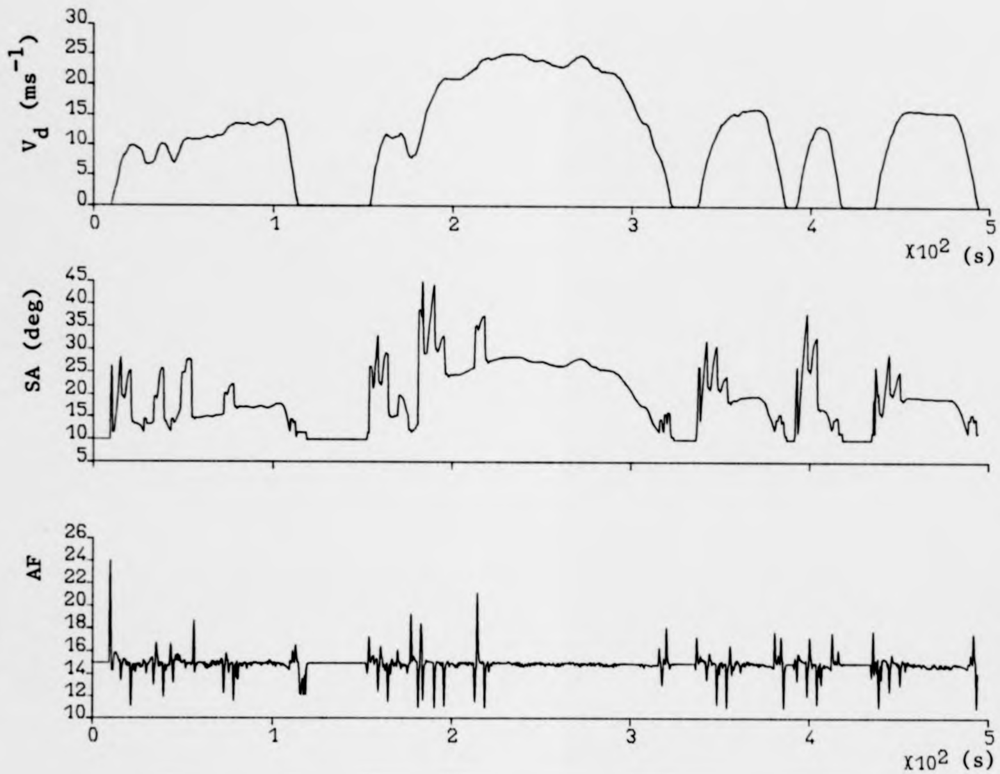

Figure 6.1 Driving schedule and engine controls for a reference run

```
                SUMMARY OF RUN (reference)

File containing engine/vehicle data  &DAT2H
File containing driving schedule  VELOC
File containing control variable coeffs. REFCNTRL

Length of driving schedule. 1370 seconds.

Distance travelled (m) =          11843.668
Fuel consumed (Kg) =                  0.793 (  1.107 litres)
Emissions produced (g)
                  CO  =              71.389
                  HC  =               8.160
                  NOx =              20.299

Mean vehicle speed (m/s) =            8.645 ( 31.12 kph, 19.34 mph)
Mean fuel consumption (l/100km)       9.346 ( 30.23 mpg)

Mean exhaust emissions (g/km)
                  CO  =               6.028
                  HC  =               0.689
                  NOx =               1.714

                SUMMARY OF RUN (SA perturbation)

File containing engine/vehicle data  &DAT2H
File containing driving schedule  VELOC
File containing control variable coeffs  CNTRL1

Length of driving schedule  1370 seconds

Distance travelled (m) =          11840.938
Fuel consumed (Kg) =                  0.766 (  1.070 litres)
Emissions produced (g)
                  CO  =              70.548
                  HC  =              10.017
                  NOx =             113.364

Mean vehicle speed (m/s) =            8.643 ( 31.11 kph, 19.34 mph)
Mean fuel consumption (l/100km)       9.035 ( 31.28 mpg)

Mean exhaust emissions (g/km)
                  CO  =               7.647
                  HC  =               0.646
                  NOx =               9.574

                SUMMARY OF RUN ([illegible] perturbation)

File containing engine/vehicle data  &DAT2H
File containing driving schedule  VELOC
File containing control variable coeffs  CNTR2

Length of driving schedule  1370 seconds

Distance travelled (m) =          11842.604
Fuel consumed (Kg) =                  0.776 (  1.083 litres)
Emissions produced (g)
                  CO  =              41.395
                  HC  =               6.085
                  NOx =              25.237

Mean vehicle speed (m/s) =            8.644 ( 31.12 kph, 19.34 mph)
Mean fuel consumption (l/100km)       9.147 ( 30.89 mpg)

Mean exhaust emissions (g/km)
                  CO  =               3.495
                  HC  =               0.514
                  NOx =               2.131
```

Figure 6.2 Simulation summary for reference and perturbation runs

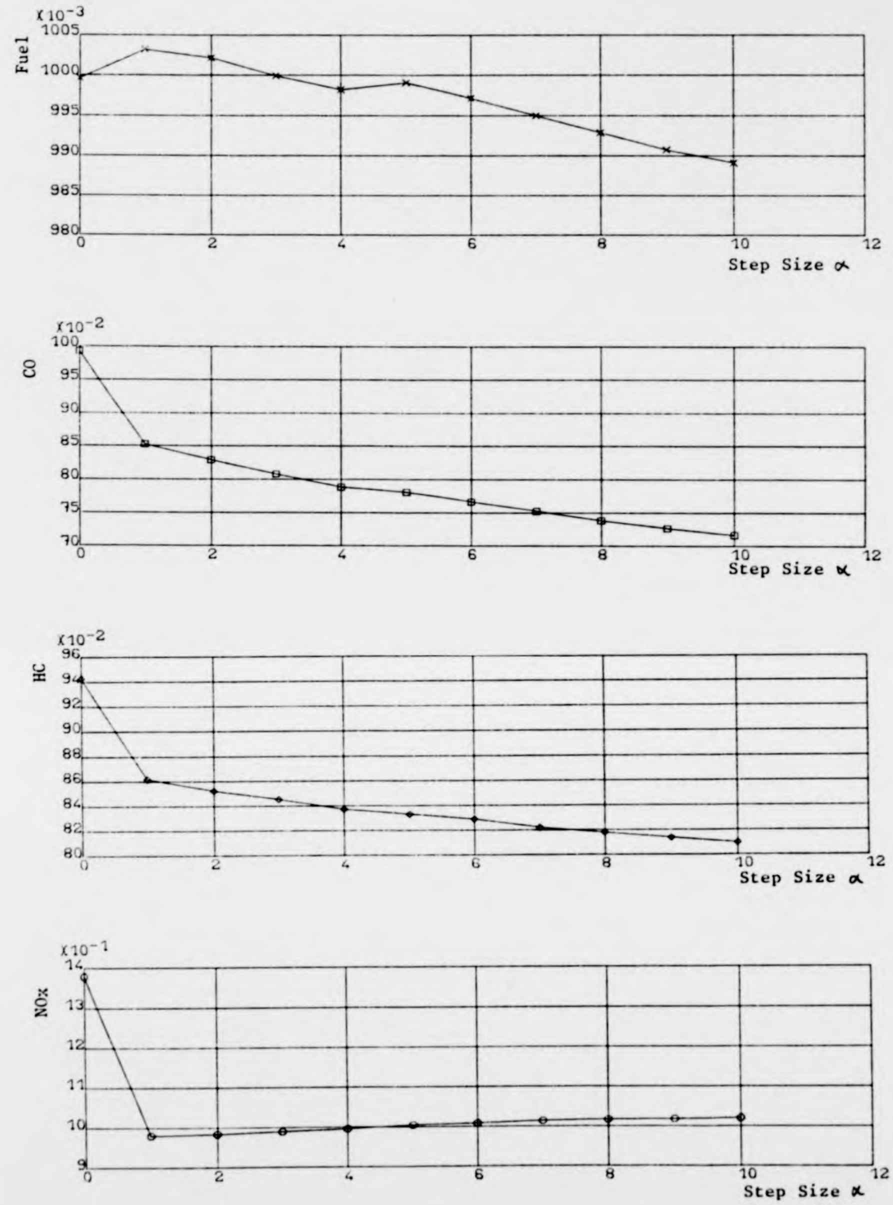

Figure 6.3 Effect of step size on first iteration
(emissions normalised to constraint)

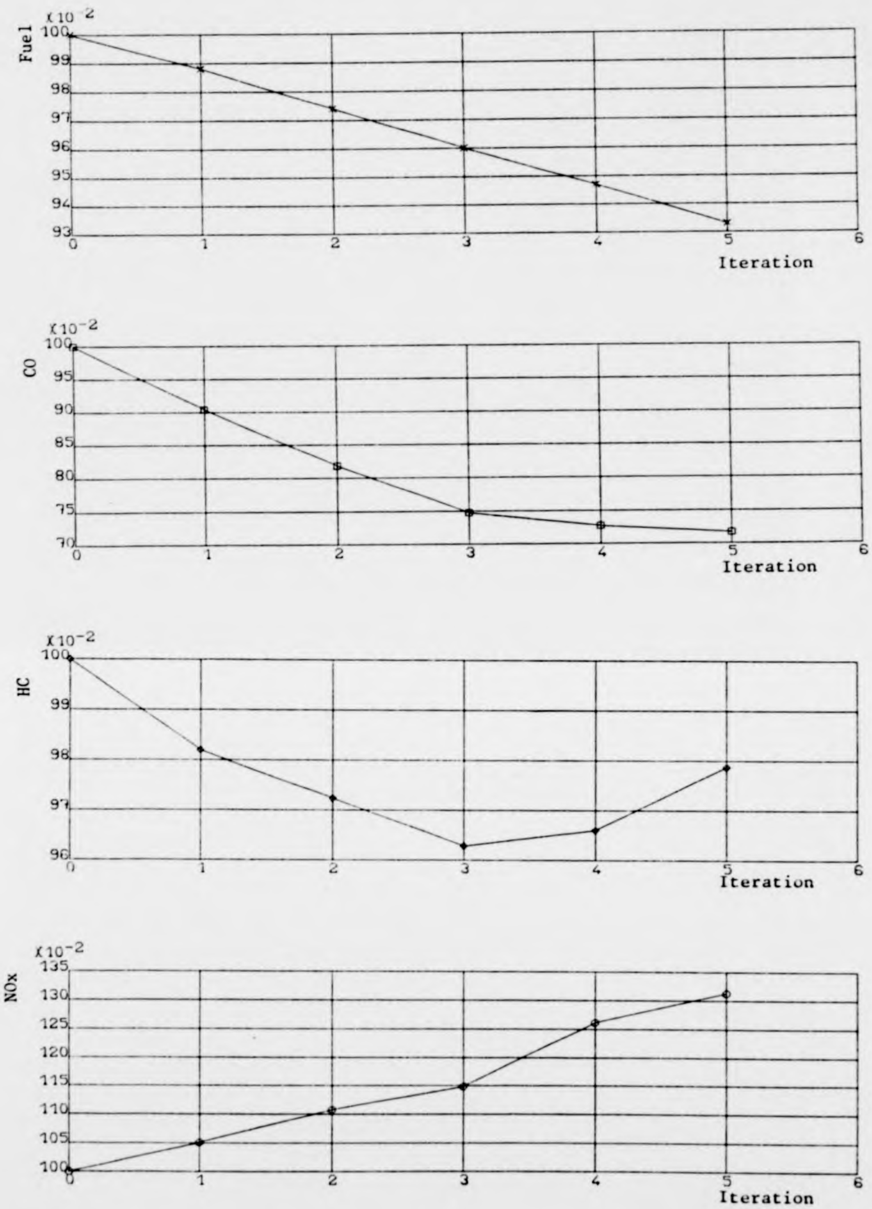

Figure 6.4 Results for unconstrained optimisation (normalised to reference run)

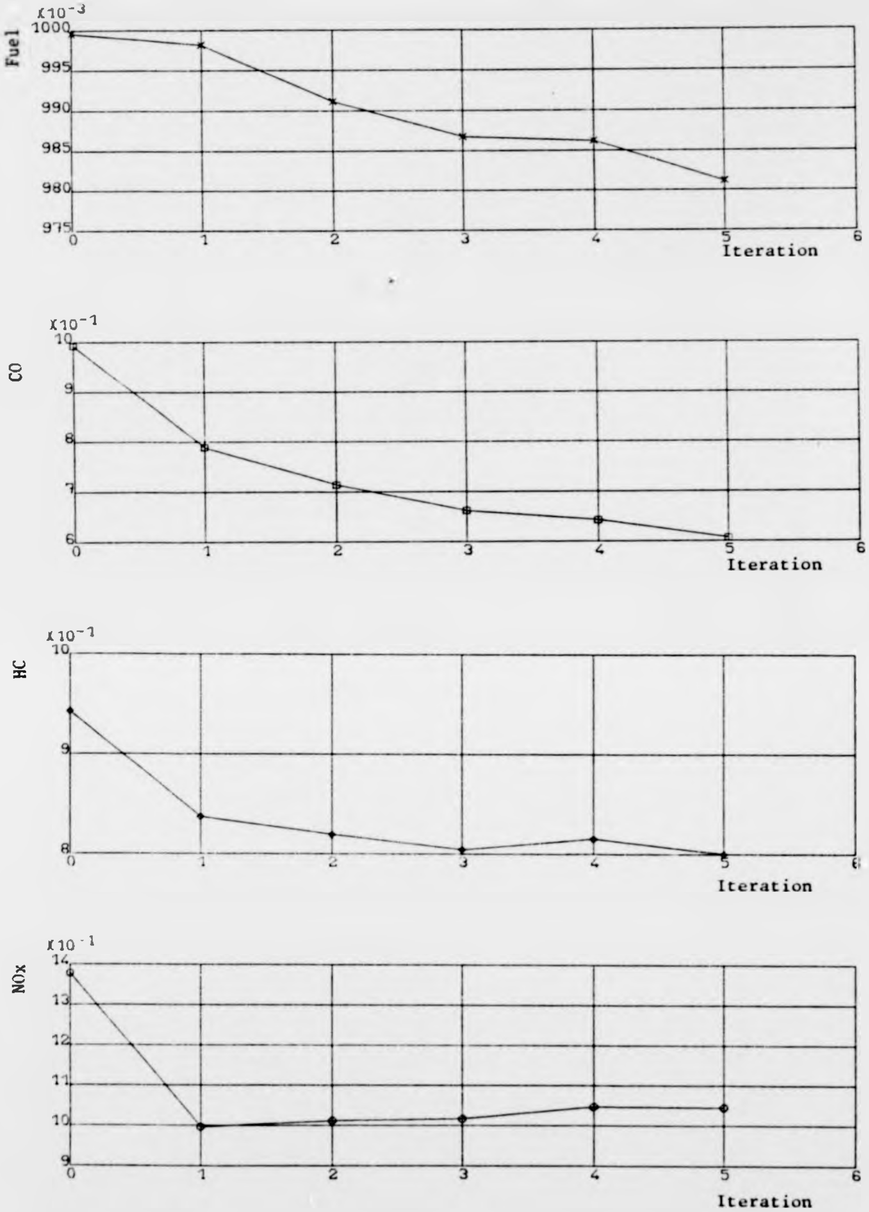

Figure 6.5 Results for a constrained optimisation (normalised to constraint)

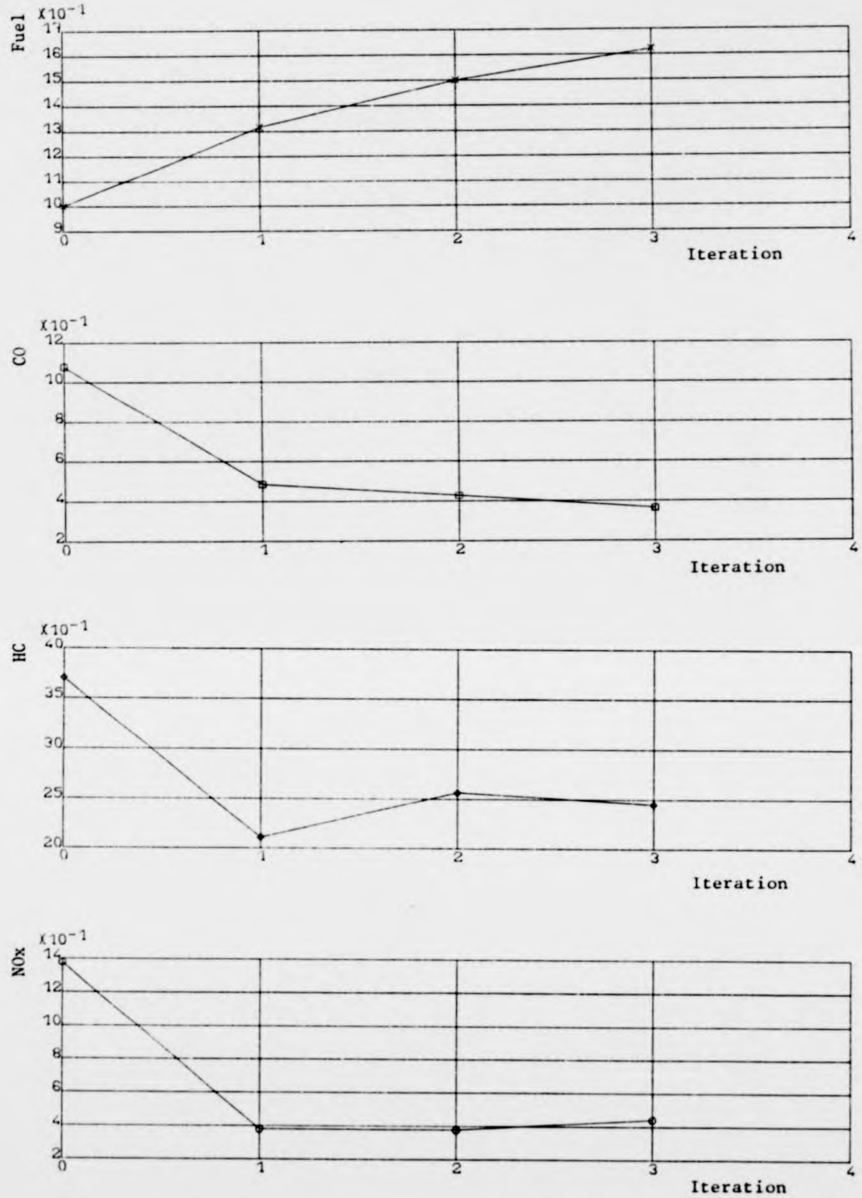

Figure 6.6 Results for optimisation with severe constraint (normalised to constraint)

CHAPTER 7

SIMULATION STUDY OF A SIMPLE ADAPTIVE ENGINE CONTROL

7.1 INTRODUCTION.

In Chapter 2 various approaches to engine control were mentioned. Predominantly on cars equipped with automatic means of controlling air-fuel ratio and ignition timing, the controls have been open-loop with feedforward compensation. The performance of these systems relative to some perceived optimum, depends on the ability of the designer to incorporate the complex functional relationships required to match the input adjustments to the output. The big attraction of an adaptive controller such as the 'Optimaliser' introduced by Draper and Li (7.1), lies in its ability to adjust its input settings based directly on the system output; without requiring detailed characterisation of the engine, and without making assumptions about environmental and other stochastic phenomena.

The simulation study reported in this chapter relates to a simple extremum-seeking ignition timing controller, similar in principle to the mechanical system used by Draper and Li.

7.2 BASIS OF A SIMPLE ADAPTIVE IGNITION CONTROLLER.

7.2.1 Extremum-seeking Controllers.

When a dynamic system incorporates some measurable characteristic that has a unique maximum (or minimum), and the object of the controller is to maintain the system at or close to that optimum; then an 'extremum-seeking' or 'hill climber' type of regulator is a possible candidate for the task. Such a controller evaluates, in some manner, the slope of the characteristic and endeavours to operate at the region of zero slope. More sophisticated controllers of this type may calculate a 'performance index' from measured variables, and manipulate the system inputs to optimise this.

At a particular constant engine speed and fueling the torque characteristic of the conventional spark ignited engine exhibits a maxima with respect to throttle angle, as shown in Figure 7.1. This Minimum angle for Best Torque (MBT) ignition timing advances at higher engine speeds and lighter loads. The controller shown schematically in Figure 7.2 is intended to track the MBT ignition timing, and thus give the lowest specific fuel consumption (sfc) with respect to this control variable.

7.2.2 A Perturbation Ignition Timing Controller.

Suppose at some steady engine speed the engine torque T, with respect to ignition advance θ , may be represented by the equation (Figure 7.2)

$$T = a - b(\theta^* - \theta)^2 \qquad (7.1)$$

where a, b are positive constants

θ^* is the MBT spark advance for this engine condition

and θ is the actual spark advance

This is reasonable as in the neighbourhood of the optimum the characteristic may be approximated by a quadratic. The functioning of the controller may then be understood by the following simple analysis:

A sinusoidal perturbation of amplitude δ and angular frequency ω is added to the spark advance setting θ_o , giving

$$\theta = \theta_o + \delta \sin\omega t \tag{7.2}$$

This is then modulated by the engine characteristic (cost function), giving

$$\begin{aligned} T &= a - b(\theta^* - \theta_o - \delta\sin\omega t)^2 \\ &= a - b[x_o - 2x_o\delta\sin\omega t + \frac{\delta^2}{2}(1 - \cos2\omega t)] \end{aligned} \tag{7.3}$$

by substitution of

$$\frac{1}{2}(1 - \cos2\omega t) = \sin^2\omega t$$

where $x_o = \theta^* - \theta_o$

The high pass filter removes the d.c. component to give

$$P = b\delta(2x_o\sin\omega t + \frac{\delta}{2}\cos2\omega t) \tag{7.4}$$

which after multiplication with the perturbation signal is

$$Q = b\delta^2[x_o(1 - \cos2\omega t) + \frac{\delta}{4}(\sin3\omega t - \sin\omega t)] \tag{7.5}$$

and after smoothing by the low pass filter provides a mean component

$$\dot{\theta}_o(t) = Kb\delta^2(\theta^* - \theta_o(t)) \qquad (7.6)$$

which is proportional to the slope of the cost function (7.1), and has the solution

$$\theta_o(t) = \theta^* - (\theta^* - \theta_o(0)) \exp[-Bt] \qquad (7.7)$$

where $B = Kb\delta^2$, and $\theta_o(0)$ is the value of $\theta_o(t)$ at time $t = 0$

This is inherently stable for all $B > 0$, under our assumption that the torque characteristic is essentially parabolic.

The controller thus measures the effect on the system output, of the control perturbation, in order to adjust the control in a direction which is most likely to increase the amplitude of the output (cost function).

7.3 SIMULATION OF THE IGNITION TIMING CONTROLLER USING CASS.

7.3.1 CASS Implementation Considerations.

Modifications to CASS

With the implementation of an adaptive ignition timing controller on a vehicle there is unlikely to be any need to modify any other part of the automotive system; there are however additional considerations when this type of control system is incorporated into a simulation model such as CASS. Apart from the normal factors to take into account with digital simulation, there may be a number of minor modifications to the simulation program code; in the case of CASS these mainly relate to the principle of operation of the driver module.

The philosophy behind the design of the driver in CASS (Section 4.6.2), is that an experienced driver rapidly learns how a particular vehicle will respond at any given speed to the driver controls (throttle angle, gear selected and clutch position) to give the desired propulsion. In the implementation of the CASS driver 'knowledge' is inherent; translating the idea of the driver having learned the vehicle response, into the ability to predict the steady state engine torque characteristics, clutch transfer characteristics, and power-train inertia effects. The engine steady state torque model is however based upon a map of brake torque versus throttle angle, engine speed, air-fuel ratio and spark advance. The driver algorithm in determining the torque for any given operating condition, must first fetch the engine calibration (air-fuel ratio, spark advance) for that condition, from the fixed control laws or map. Evidently, when the ignition timing is controlled in an adaptive manner, there is no way of predicting what the spark advance will be at a particular operating condition. Thus there is a need to modify the driver algorithm.

In CASS the main modifications to the driver, for the purpose of simulating the perturbation ignition timing controller, were

a) in the gear selection algorithm assume that the MBT ignition timing will be achieved. Thus in determining the maximum or minimum torque at a particular engine speed, the MBT spark advance is used (evaluated from the torque model).

b) when setting the throttle angle to give the desired torque (Section 4.6.5) at a particular instant, the nominal ignition advance (i.e. without the perturbation) is used.

Early simulation tests of this system revealed the need for an additional modification. Owing to the fact that the CASS driver has closed loop control over the vehicle speed, based on 'knowledge' of the engine torque characteristic, it was found that the driver was attempting to compensate for the control perturbations. The throttle actuator incorporated a rate limit on throttle angle, but no other dynamics; it was therefore decided to add a 0.25s first-order lag to the actuator: for the 0.05 second time step of the simulation program main loop this was implemented as

$$\phi_{t_k} = 0.8187\ \phi_{t_{k-1}} + 0.18127\ u_{t_k} \tag{7.8}$$

where

ϕ_{t_k} is actual throttle angle at time t_k

$\phi_{t_{k-1}}$ is the throttle angle at the previous time step

u_{t_k} is the 'desired' throttle angle at time t_k

Calculation of MBT Ignition Timing

In order to assess the ability of the controller to follow the minimum spark advance for best torque (MBT) it was necessary to calculate this following a simulation. The engine steady state torque model is a function of engine speed, air-fuel ratio, throttle angle and spark advance, and at discrete intervals these variables were logged. The torque function was then used to obtain the MBT timing for comparison with the logged value.

For a given engine speed, throttle angle and air-fuel ratio the torque-spark advance characteristic is unimodal (Figure 7.1), not monotonic like the torque throttle angle characteristic (Figure 4.11). This means that the method of bisection, used for obtaining 'desired' throttle angle (Section 4.6.5), is unsuitable for the present purpose; instead a direct search method was used.

The range of search was restricted to [0, 80] degrees c.a., and a fixed step size was used; beginning the search from the upper limit and evaluating the torque function at each step using the assumption of unimodality until the maximum has been bracketed. A reduced step size was then used over the sub-interval until the maximum was again bracketed. The initial step size was 10°, reducing to 1° and then to 0.1° in order to obtain the MBT ignition timing to within ±0.05°.

A fixed step size method such as this generally is not very efficient, but it does have the advantages of being very simple to programme and of numerical stability. In this instance where the range of search is small, and computational efficiency is not important, the method is adequate. Under other circumstances alternative elimination methods (e.g. Golden Section method) or interpolation methods will prove more suitable, if rather more difficult to implement (7.2).

Controller Input Specification

Nominal air-fuel ratio is controlled as a programmed function as previously (Section 5.5.6); but necessarily, the ignition timing control requires not model coefficients, but other specification parameters:

i) static (idle) spark advance (degrees c.a.)
ii) dither amplitude (degrees c.a.)
iii) period of advance perturbation (s)
iv) period of retard perturbation (s)
v) gain constant for the integrator
vi) maximum allowable advance (degrees c.a.)

Parameters (i) and (vi) are the minimum and maximum bounds on spark advance, necessary to help ensure operation within the bounds of the engine model. The period of advance and retard perturbation (iii - iv) may be different, and has application when a rectangular perturbation is used and a bias in one direction is to be introduced. The higher the gain (v) of the integrator the more sensitive the controller will be to perceived deviations from the optimum ignition timing.

Controller Implementation

The function of the controller broadly follows that shown in Figure 7.2, with the incorporation of a delay in the perturbation signal to the multiplier, in order to match the phase lag due to the engine and high pass filter. In addition the perturbation could be chosen as rectangular or sinusoidal, and both upper and lower bounds on nominal ignition advance were imposed.

Both filters were tan Butterworth types with appropriate cut-off frequencies. As well as setting the value of spark advance the programme provided output to a separate file, of the intermediate calculations - the measured torque, the perturbation, and the outputs of the multiplier, both filters and the integrator; these were used in analysing the performance of the controller.

7.3.2 Simulated Behaviour of the Perturbation Controller.

Selection of the response (measured) variable

Engine speed information, whether obtained from the flywheel or distributor, has been used in a number of applications requiring the measurement of variations in engine torque (7.3-7.5). This is chiefly due to the low cost, robustness and reliability of sensors for speed or acceleration measurement, when compared to the problems of direct torque measurement. In the development of the simulation model of the controller for use in CASS engine speed was initially chosen as the variable to be maximised; it did however prove unsuitable.

The aim of this extremum-seeking control is to minimise the specific fuel consumption by adjusting the nominal spark advance in a direction which maximises the engine power. Evidently an advantageous perturbation of spark advance will tend to increase engine torque, accelerating the flywheel; thus it should be possible to correlate the measured speed of the flywheel with the ignition timing perturbation signal. Though this may seem reasonable for a vehicle equipped with a fluid flywheel or torque converter, it does not follow that it is feasible for a vehicle with a manual transmission on account of the large difference in effective compliance of the coupling between the engine flywheel and the remainder of the powertrain, between the two systems.

The system represented in this work was of a manual transmission, and it proved difficult under simulation conditions to retrieve the desired information from the engine speed. This situation is perhaps not surprising considering a non-compliant powertrain model is being used in the simulation, but it is not unlikely that similar difficulty may be encountered on an actual vehicle system.

As the controller being studied was intended to adjust ignition timing in a direction tending to maximise engine torque, it seemed reasonable then to use the simulated torque as the 'measured' variable. Variations in torque may in practice lead to variations in flywheel speed, but the flywheel inertia is designed to help remove unwanted and rapid torque fluctuations due to cyclical combustion phenomena. The direct measurement of engine torque on a vehicle poses some difficulty, although in fact it is the mean engine pressure which is causal, and this can in some sense be measured.

Choice of perturbation frequency

Owing to the discrete nature of the internal combustion engine cycle the ignition timing setting is effectively sampled at regular intervals; for a four cylinder 4-stroke engine running at 1000 r.p.m. the sampling rate is approximately 33 Hz. This would place an upper limit on perturbation frequency of around 16.5 Hz, but in practical systems it would be advisable to choose a frequency well below this.

As CASS was set up, the step size of the main program loop was 0.05 seconds, but this is not in fact synchronised with engine firing as a discrete combustion model was not being used. As the Nyquist folding frequency is now at 10 Hz, it was decided on the basis of some limited simulation experiments, to use a sinusoidal perturbation with period 0.3 seconds for most of the studies.

The simulated performance of the controller in fact was very little affected by increasing the perturbation frequency, as long as the filters were carefully chosen; in particular there seemed to be no improvement to be gained by using a 5 Hz sinusoidal signal in favour of the 3.33 Hz. Neither did there appear to be a significant difference between a controller using a sinusoidal perturbation and one using a rectangular perturbation of the same frequency; this can be attributed to the fact that considerable attenuation of the harmonics will occur in the system.

Choice of filter characteristics

Figure 7.2 shows the presence of two filters in the controller; a high-pass and a low pass filter. The characteristics of these filters have an important role in determining the overall performance of the controller.

The high pass filter ideally removes the low frequency content in the measured variable, below the perturbation frequency, and allows all other information to pass unattenuated. Conversely an ideal low-pass filter removes all frequency components in the signal above a certain frequency threshold known as the 'band-width'. Such ideal filters would require an infinite number of terms to implement perfectly, so in practice they are approximated using polynomials.

Frequently Butterworth filters are used in engineering as they can provide a good approximation to the ideal form. For the analogue low pass filter a common form is

$$|H(f)|^2 = \frac{1}{1 + (2\pi f/2\pi B)^{2P}}$$

where H(f) is the transfer function of the filter

B is the 'bandwidth'

P is a positive integer

In the digital case Butterworth filters are trigonometric functions of f; the tan form is

$$|H(f)|^2 = \frac{1}{1 + (\tan\pi fT/\tan\pi BT)^{2P}} \qquad (7.9)$$

for the low pass filter, and for the high pass filter the tan form is

$$|H(f)|^2 = \frac{1}{1 + (\cot\pi fT/\tan\pi BT)^{2P}} \qquad (7.10)$$

where T is the sample period. A similar form is possible using sine instead of tan.

The characteristics of the LP filter of equation 7.9 are easily determined:

$$|H(f)|^2 = \begin{cases} 1 , & f = 0 \\ 1/2 , & f = B \\ < 1/2 , & f > B \end{cases}$$

The higher the value of P the better the approximation, but the higher the computational requirement. The tan Butterworth filter characteristics for different values of P are shown in Figure 7.3 for a HP filter (B = 2.5 Hz) and a LP filter

(B = 5 Hz); in both cases the sampling period is 0.05 seconds, giving a Nyquist folding frequency of 10 Hz.

Apart from being among the simplest higher order filters to implement Butterworth filters are widely applicable for a number of reasons. The gain of the filter tends to be flat in the passband until the frequency approaches B, when it rapidly becomes small and remains virtually flat in the stopband (Figure 7.3). In addition the phase characteristic is almost constant across the passband, thus approximating to a pure delay.

For the application in the extremum seeking controller it was essential to have a sharp cut-off characteristic for the high pass filter, as the perturbation frequency (3.33 Hz) was close to the required stopband. The choice of bandwidth and order (P) of the filters was made with reference to the power spectra of the signals at various points in the system (Figure 7.4). For this particular perturbation frequency a 6-pole HP filter was chosen, with the -3dB attenuation at 2.5 Hz; and a 4-pole LP filter with a 5 Hz bandwidth - both of which are shown in Figure 7.3.

The filters were implemented recursively using a difference equation of the form

$$y(i) = \sum_{k=0}^{K} b_k x(i-k) - \sum_{p=1}^{P} a_p y(i-p) \tag{7.11}$$

to give the output y(i) by linearly combining K past inputs and the current input x(i), with P past outputs; the recursive filter weights $b_0 \ldots b_K$, $a_1 \ldots a_P$ first having been calculated. In general it is the number of previous output terms employed which determines the order of the filter. In particular if the transfer function H(f) of the sine or tan filter was expressed as a rational polynomial, there would be P poles (denominator roots) giving a pth order filter. The sine filter has no zeros (numerator roots) and thus K = 0 in the expression 7.11; this means that the tan Butterworth filter has a higher computational cost (as it has zeros) than the sine filter, but it does possess a much better attenuation than the latter in the stopband.

The computation of the filter weights is fully described by Otnes and Enochson in reference 7.6, where a Fortran programme is provided for this purpose. A cascade implementation is employed, using two or three 2nd order stages. The coefficients and Fortran code for these filters are shown in Table 7.1 and Figure 7.4 respectively.

TABLE 7.1(a)

Coefficients for the LP Digital Butterworth Filter

Stage	a_1	a_2	b_0	b_1	b_2
1	0.0	0.03957	0.30656	0.61313	0.30656
2	0.0	0.44646	0.30656	0.61313	0.30656

TABLE 7.1(b)

Coefficients for the HP Digital Butterworth Filter

Stage	a_1	a_2	b_0	b_1	b_2
1	-0.84029	0.18835	0.59271	-1.18542	0.59271
2	-0.94281	0.33333	0.59271	-1.18542	0.59271
3	-1.19543	0.69060	0.59271	-1.18542	0.59271

General characteristics of the controller

A fuller appreciation of the analysis of Section 7.2.2 and the function of the filters, may be obtained by examining the frequency domain characteristics of the controller. The single-sided power spectra of Figure 7.5 were obtained from a simulation run over the EPA Highway driving schedule; using the filters described above, an integrator gain of 0.2 (relative) and a sinusoidal perturbation, amplitude 3.0 degrees.

The high-pass filter effectively removes the large d.c. component from the torque 'measurement' (7.5a), leaving the information contained at and above the perturbation frequency virtually unattenuated; as expected there is a significant component close to the perturbation frequency (7.5b).

The multiplication of the perturbation signal with the filtered torque signal in the time domain, is equivalent to the convolution of Figure 7.5c and 7.5b (frequency domain).

The frequency spectra of the sinusoid (Figure 7.5c) ideally would be a 'spike'; its width here is due to the sampled nature of the data, the finite length of the data record, and the resolution of the frequency axis. The convolution can be seen to result in a large low frequency component and a smaller double frequency component (Figure 7.5d), which are readily understood from considering the convolution of the two-sided spectra. The low-pass filter at 5 Hz smooths the signal removing the double frequency component (7.5c), which after integration results in the nominal spark advance having significant power at zero frequency.

The EPA Highway driving schedule (Figure 7.6), which formed a basis for the frequency responses illustrated in Figure 7.5, is 12.7 minutes long. A single simulation will thus produce 15,240 data samples recorded at 0.05 second intervals (simulation time); for the 40 variables recorded this results in more than 2 Mbyte data per test. It is, however, desirable to use a data record which is as long as practical in order to reduce the effects on the frequency characteristics of the discontinuities at either end of the data record.

This driving schedule is also used below to illustrate the controller characteristics in the time domain; it is particularly useful for studying the controller performance, as it includes a fairly lengthy cruise condition as well as rapid transients such as characterise most urban schedules.

Generally it was expected that the speed of response of the controller to changing operating conditions would be slow, but its capability of achieving MBT ignition timing under near steady conditions would be unsurpassed. The simulation results in Figure 7.7 and 7.8 clearly bear out these expectations: here the perturbation amplitude is 3.0°, and a gain constant (relative) of 0.10 is used. Figure 7.7 is for a region of the highway schedule which involves rapid changes in engine operating condition, while Figure 7.8 represents a period of operation with considerably less variation in conditions.

Effect of perturbation amplitude

From equation 7.7 the gain of the controller can be seen to be proportional to the square of the perturbation amplitude. Direct assessment of the effect of modifying the perturbation amplitude can thus be facilitated by a commensurate adjustment of the gain K (equation 7.7) for a repeat simulation.

A perturbation amplitude of 1.0° and a gain K of 0.9 was used for comparison with the simulation which gave rise to Figures 7.7 and 7.8. In the results illustrated in Figure 7.9, the response of the two controllers is very similar; the chief difference lies in the sensitivity of the second controller to spurious information in the torque signal. This is due to any torque changes not caused by the perturbation, but having significant power close to the perturbation frequency. The relatively large difference in spark advance between the two controllers at the beginning of the interval shown in Figure 7.9, is due to a gear change occurring immediately prior to the interval - a rapid transient for which no setting of the controller proved able to cope with adequately.

The conclusion resulting from a number of simulations involving different perturbation amplitudes, was that an amplitude of 3° was sufficiently small to support an assumption of linearity in the vicinity of operation, but not so small as to encounter serious problems due to a poor signal to noise ratio.

<u>Effect of controller gain</u>

From equations 7.6 and 7.7 it is evident that the transient response of the system is limited by the parameter b (of our approximation to the torque characteristic), the perturbation amplitude as discussed above, the relative gain constant K, and the phasing between the perturbation components applied to the multiplier. The perturbation frequency does not have a significant effect, except that it must be chosen to avoid frequencies with a significant noise content.

The effect of adjusting K is revealed in Figures 7.10 to 7.13, where K has values 0.05, 0.10, 0.20, 0.30 respectively. Evidently, as expected, the transient response is faster with a higher gain, but an oscillation of about 0.35 Hz is evident for a K of 0.20 (Figure 7.12b) and increases in amplitude as the gain increases (Figure 7.13b); this is not represented in the linear analysis above but is excited by the ripple in the torque, giving rise to a corresponding ripple in the MBT spark advance. The system gain K is the only parameter of this simple controller which may effectively be altered, and any increase in K has no effect on the damping of the system, resulting in an increased tendency towards oscillation under certain circumstances.

As indicated above, the phasing of the two components applied to the multiplier have an important effect on the controller gain; the gain is a maximum when the two signals are in phase, falling to zero for an angle of $\pm\pi/2$ when

$$Q = b\delta^2[x_0\sin 2\omega t + \frac{\delta}{4}(\cos 2\omega t - \cos 3\omega t)] \qquad (7.12)$$

(cf. equation 7.5)

If the angle is $\pm\pi$ then a change of sign occurs causing the spark advance to process away from the MBT condition; if the two multiplier components are in antiphase then

$$Q = -b\delta^2[x_0(1 - \cos 2\omega t) - \frac{\delta}{4}(\sin 3\omega t - \sin\omega t)] \qquad (7.13)$$

and

$$\dot{\theta}_0(t) = -Kb\delta^2(\theta^* - \theta_0(t)) \qquad (7.14)$$

(cf. equation 7.6)

7.4 CONCLUSIONS.

An extremum-seeking controller as described in this chapter has considerable potential advantages over its open-loop counterpart, if the target is the minimum ignition timing angle for best torque (MBT). These advantages in particular are: the need to sense only one variable, the insensitivity to sensor accuracy, the ability to respond to unsensed or stochastic variables, and the low total cost of the system.

Weighted against this is one disadvantage, so important that it renders the controller infeasible, in this simple form, for use on a normal road vehicle; this disadvantage is its slow speed of response, as is borne out by the simulations described in Section 7.3.2. The simulations showed that its steady-state performance was excellent, but under normal transient driving conditions as described by the EPA Urban and Highway schedules, its performance at best would be unacceptable.

The controller referred to here is intended to select the optimum spark timing with respect to fuel economy. It is not clear at first that adjusting the spark timing in a direction tending to increase engine power will necessarily bring this about. The driver, however, is an important element in the system, as for some desired steady speed and load, he will reduce the throttle angle (and hence fuel consumption) as the spark timing processes towards MBT. The measurement of the engine variable to be maximised can of itself be a significant problem: the output power cannot be measured directly, but flywheel speed can easily be measured and regularly increases and decreases with engine power; in the same way torque at the flywheel and IMEP are suitable candidates for the extremum control system.

While acceleration/deceleration of the flywheel has been used with vehicles equipped with automatic transmissions (7.3), flywheel speed was an unsuitable measurand for the vehicle with manual transmission and non-compliant drive-train represented using CASS; where the effective inertia at the flywheel severely attenuates speed fluctuations in the region, or above, the perturbation frequency. A compliant representation of the drive train possibly would allow the use of flywheel speed, but would not approach the measurement conditions due to the use of a torque converter or fluid flywheel. From the flywheel speed measurement aspect, a drive-train with a lock-up torque converter (in the locked mode) would exhibit similar characteristics to a drive-train with a normal manual transmission.

Torque at the flywheel is perhaps closer to the idea of measuring engine output power; particularly with a vehicle having a manual transmission. The main problem here is finding and installing a suitable sensor in the confined space. Changes in engine power are directly attributable to changes in mean engine pressure, which is another variable which could be used in the controller. CASS could not be used for simulation in this event as discrete combustion is not represented in the underlying engine model. In practice, however, the pressure in all combustion chambers would need measuring to achieve a sufficient data rate, due to the 'sampling' process performed by the engine on the ignition timing.

Using torque as the engine variable to be optimised, the simulation of the controller revealed excellent behaviour under near steady-state engine conditions, accurately tracking MBT spark advance. During moderate transients the ignition timing lagged the optimum, and gave a performance which would be unacceptable on an actual vehicle during rapid transients. Better transient performance is achieved by increasing the controller gain, but this was limited by the tendency towards underdamped behaviour at high levels of gain.

This simple controller, while being unsuitable for controlling ignition timing in conventional road vehicles, could prove useful in other applications where transient conditions are infrequently encountered and steady-state fuel economy, under widely differing ambient conditions, is important. Another application which the simulations have shown feasible is the incorporation into an engine test bed of a controller for rapidly finding the MBT ignition timing for steady-state engine conditions; this could then provide the basis for an engine mapping exercise, or studies relating to the effect of fuelling system changes. It would probably be necessary in such an application to incorporate some means of reducing spark advance under conditions where unacceptable engine knock would occur at MBT ignition timing.

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NOTATION

T	engine torque
a	parameter in engine torque model
b	parameter in engine torque model
θ	spark advance
θ^*	MBT spark advance
θ	nominal spark advance
x_o	deviation from MBT (ie. $\theta^* - \theta_o$)
δ	ignition timing perturbation amplitude
ω	angular frequency of perturbation
P	output of high-pass filter
Q	output of multiplier
K	gain constant
B	control loop gain
ϕ_{t_k}	throttle angle at time step t_k
$\phi_{t_{k-1}}$	throttle angle at previous time step t_{k-1}
u_{t_k}	'desired' throttle angle at time step t_k

Filter parameters:

f	frequency (Hz)
B	filter 'bandwidth'
P	positive integer
H	filter transfer function
$x(i)$	filter input, ith time step
$y(i)$	filter output, ith time step

$a_1 \ldots a_p$, $b_0 \ldots b_k$ filter weights

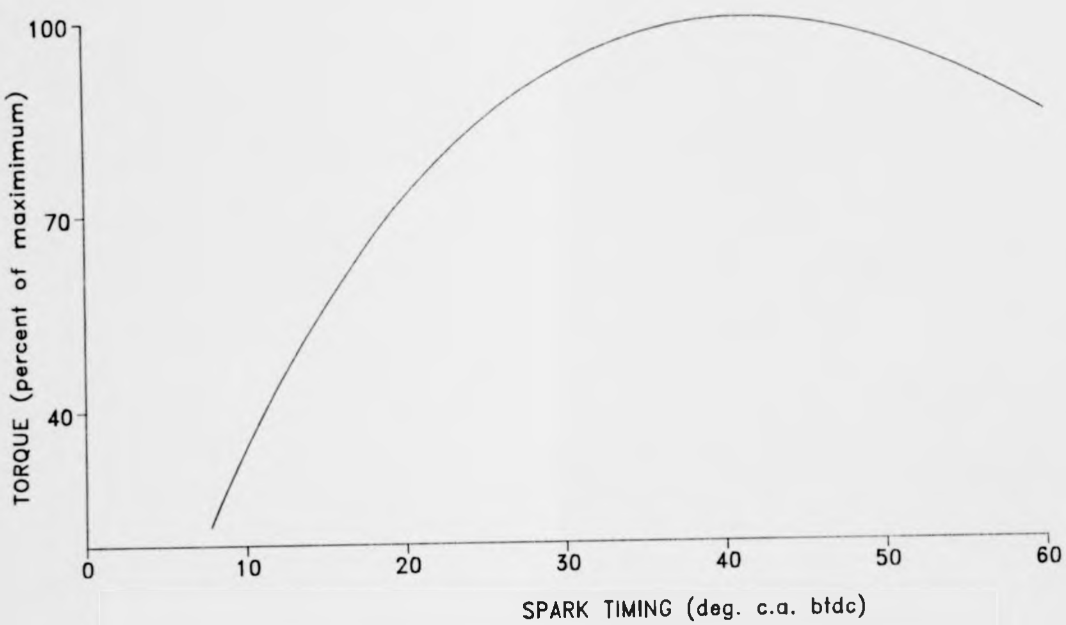


Figure 7.1 Variation of Engine Torque with Ignition Timing

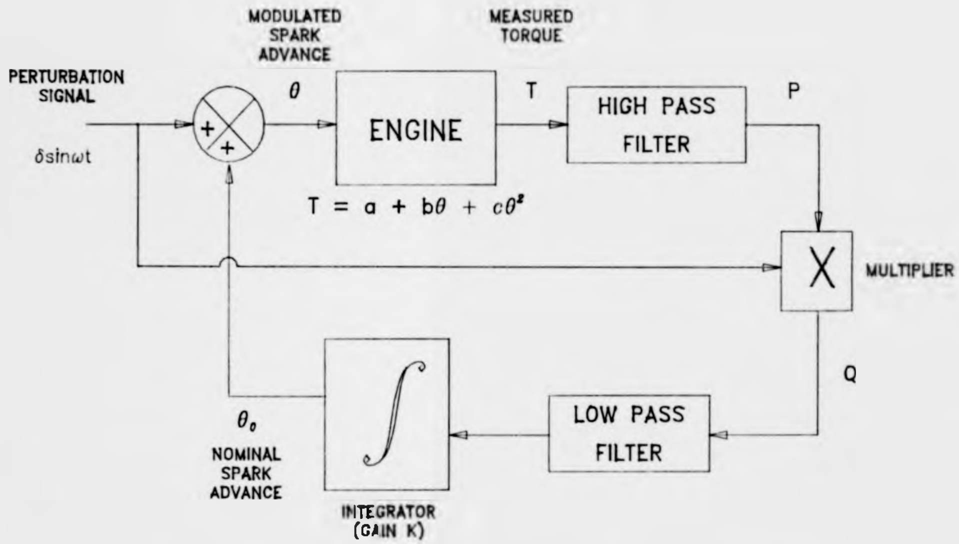


Figure 7.2 Schematic Diagram of a Simple Adaptive Ignition Timing Controller

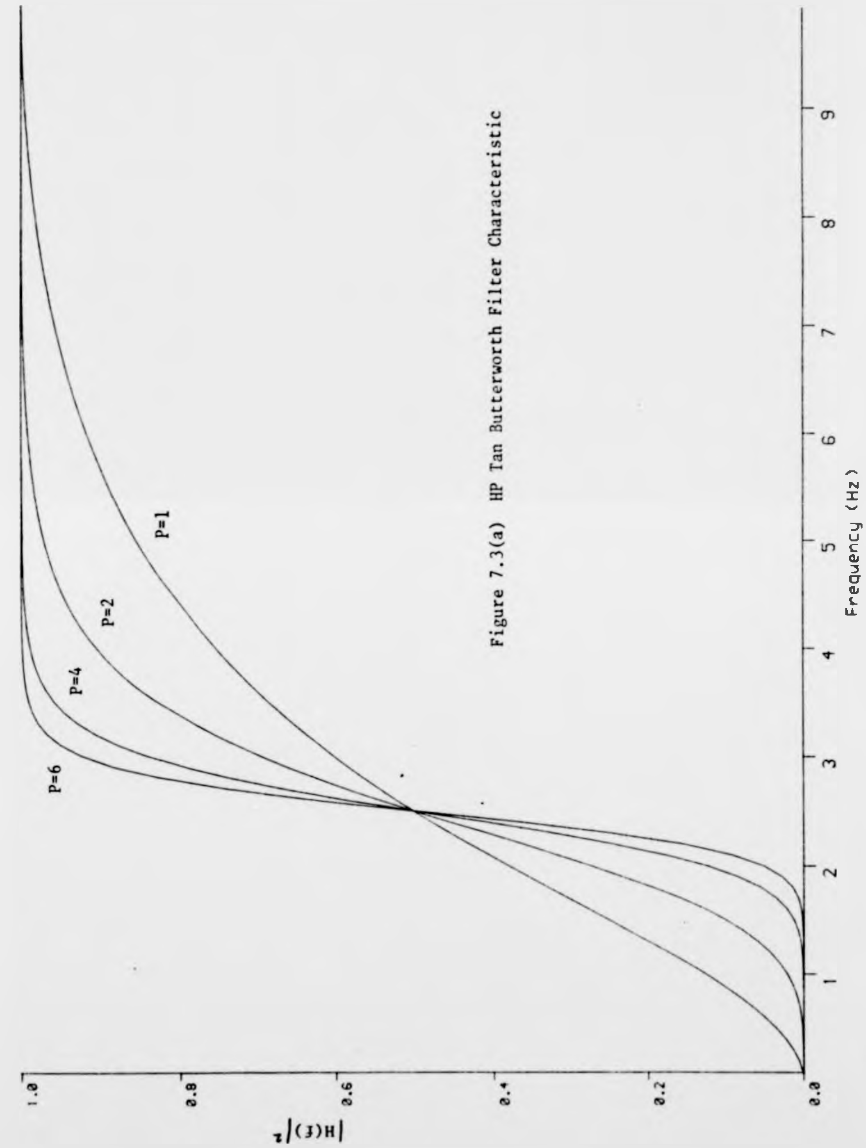


Figure 7.3(a) HP Tan Butterworth Filter Characteristic

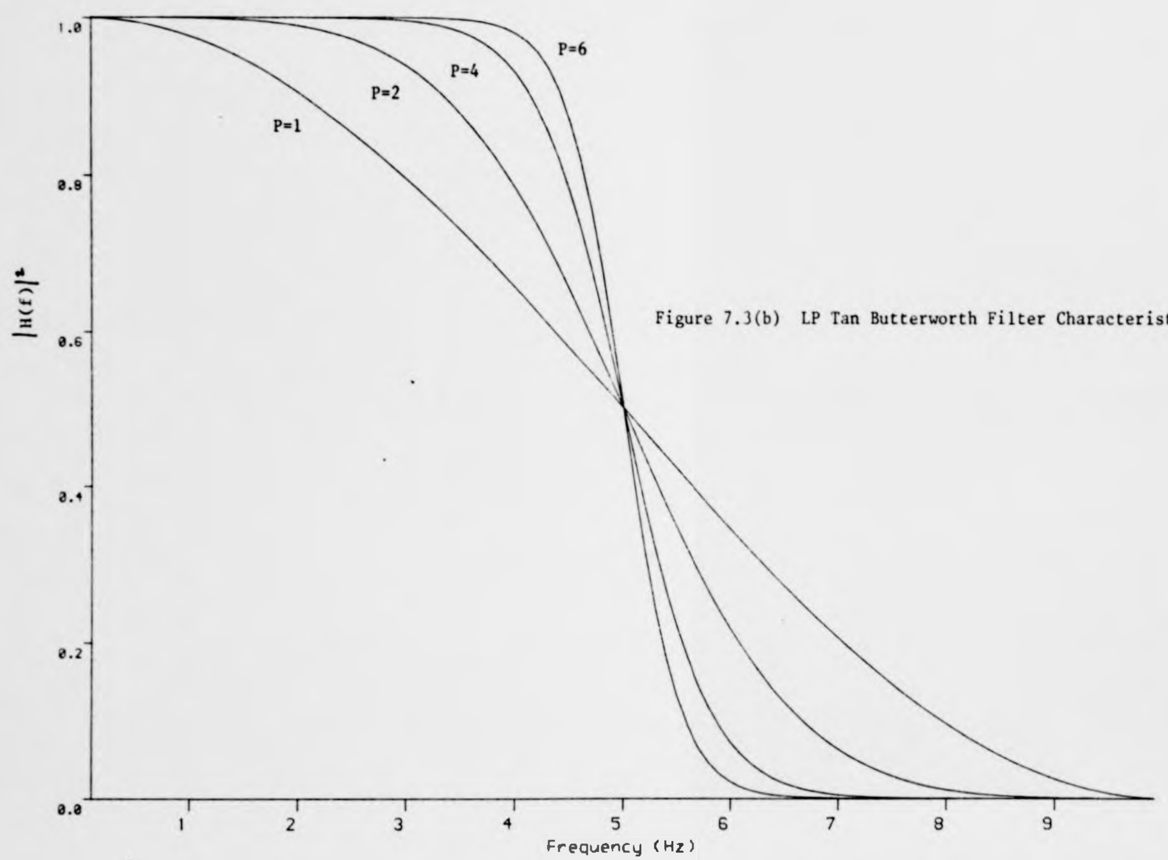


Figure 7.3(b) LP Tan Butterworth Filter Characteristic

```
C 2nd. order 6-pole Butterworth H.P. filter (2.5hz)
C Used to filter 'measurement' of engine output torque.
      REAL FUNCTION HP(XX)

C     3-stages:-
C     Y(i) = -A1*Y(i-1) - A2*Y(i-2) + B0*X(i) + B1*X(i-1) + B2*X(i-2)

$INSERT COMMON
      REAL*4 YO(4),Y1(4),Y2(4),RA1(3),RA2(3),RBO(3),RB1(3),RB2(3)
      DATA RA1,RA2,RBO,RB1,RB2/-0.84029,-0.94281,-1.19543,
     &0.18835,0.33333,0.69060,0.59271,0.59271,0.59271,
     &-1.18542,-1.18542,-1.18542,0.59271,0.59271,0.59271/

      YO(1)=XX
      DO 10 K=1,3
         K1=K+1
         YI=RBO(K)*YO(K)+RB1(K)*Y1(K)+RB2(K)*Y2(K)
     &          -RA1(K)*YO(K1)-RA2(K)*Y1(K1)
         Y2(K1)=Y1(K1)
         Y1(K1)=YO(K1)
         YO(K1)=YI
   10 CONTINUE

      Y2(1)=Y1(1)
      Y1(1)=YO(1)
      HP=YI

      RETURN
      END
```

Figure 7.4(a) Fortran code for high pass filter

```
c 2nd. order 4-pole Butterworth L.P. filter (5hz)
C Used to smooth multiplier output before integration
      REAL FUNCTION FLP(XX)

C     2-stages:-
C     Y(i) = -A1*Y(i-1) - A2*Y(i-2) + B0*X(i) + B1*X(i-1) + B2*X(i-2)

$INSERT COMMON
      REAL*4 YO(3),Y1(3),Y2(3),RA1(2),RA2(2),RBO(2),RB1(2),RB2(2)
      DATA RA1,RA2,RBO,RB1,RB2/0.0,0.0,0.03957,0.44646,
     &0.30656,0.30656,0.61313,0.61313,0.30656,0.30656/

      YO(1)=XX
      DO 10 K=1,2
         K1=K+1
         YI=RBO(K)*YO(K)+RB1(K)*Y1(K)+RB2(K)*Y2(K)
     &          -RA1(K)*YO(K1)-RA2(K)*Y1(K1)
         Y2(K1)=Y1(K1)
         Y1(K1)=YO(K1)
         YO(K1)=YI
   10 CONTINUE

      Y2(1)=Y1(1)
      Y1(1)=YO(1)
      FLP=YI

      RETURN
      END
```

Figure 7.4(b) Fortran code for low pass filter

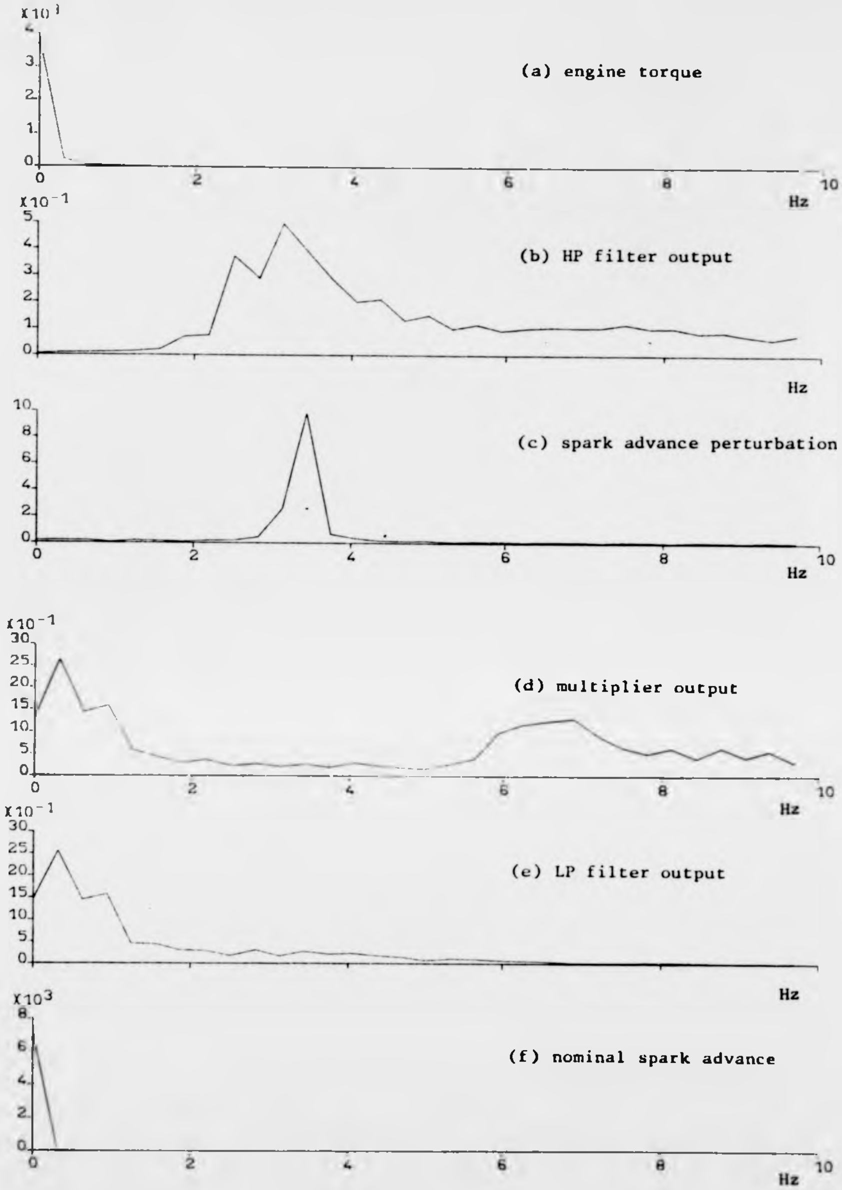


Figure 7.5 Power spectra at various parts of the controller

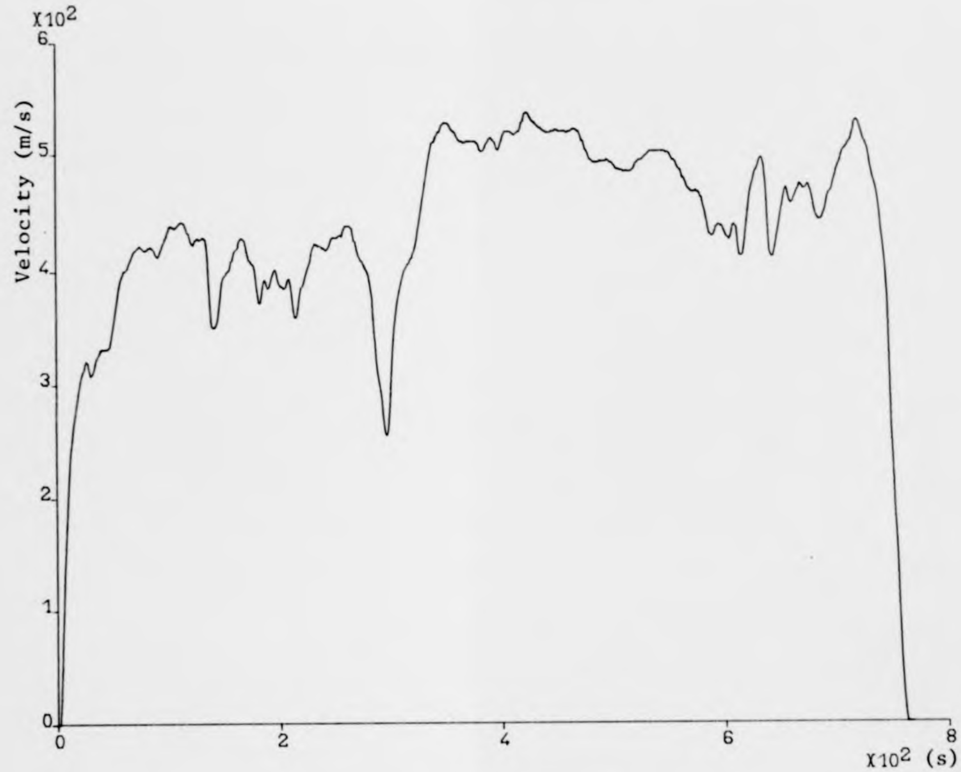


Figure 7.6 EPA Highway driving schedule

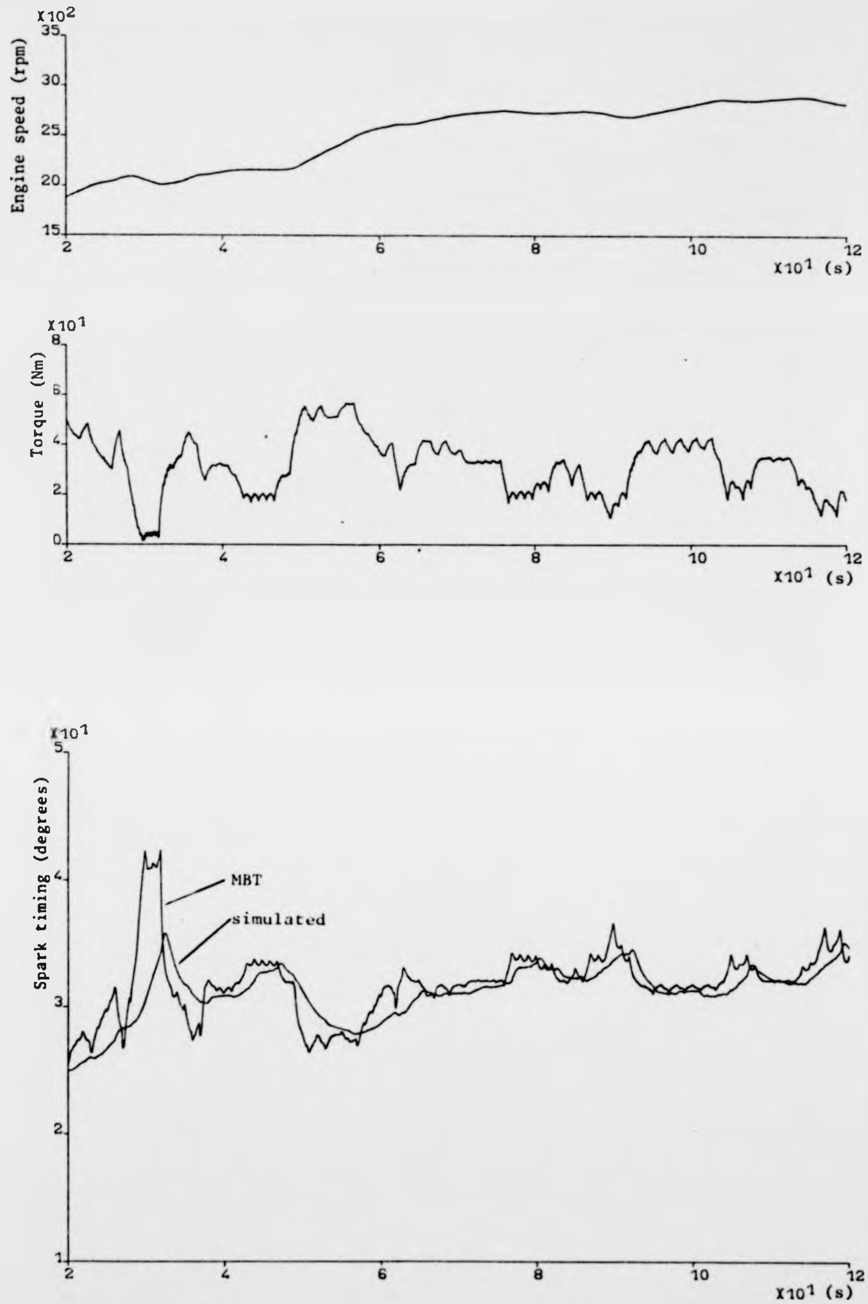


Figure 7.7 Controller performance on a portion of the EPA Highway schedule (perturbation amplitude = 3, gain K = 0.10)

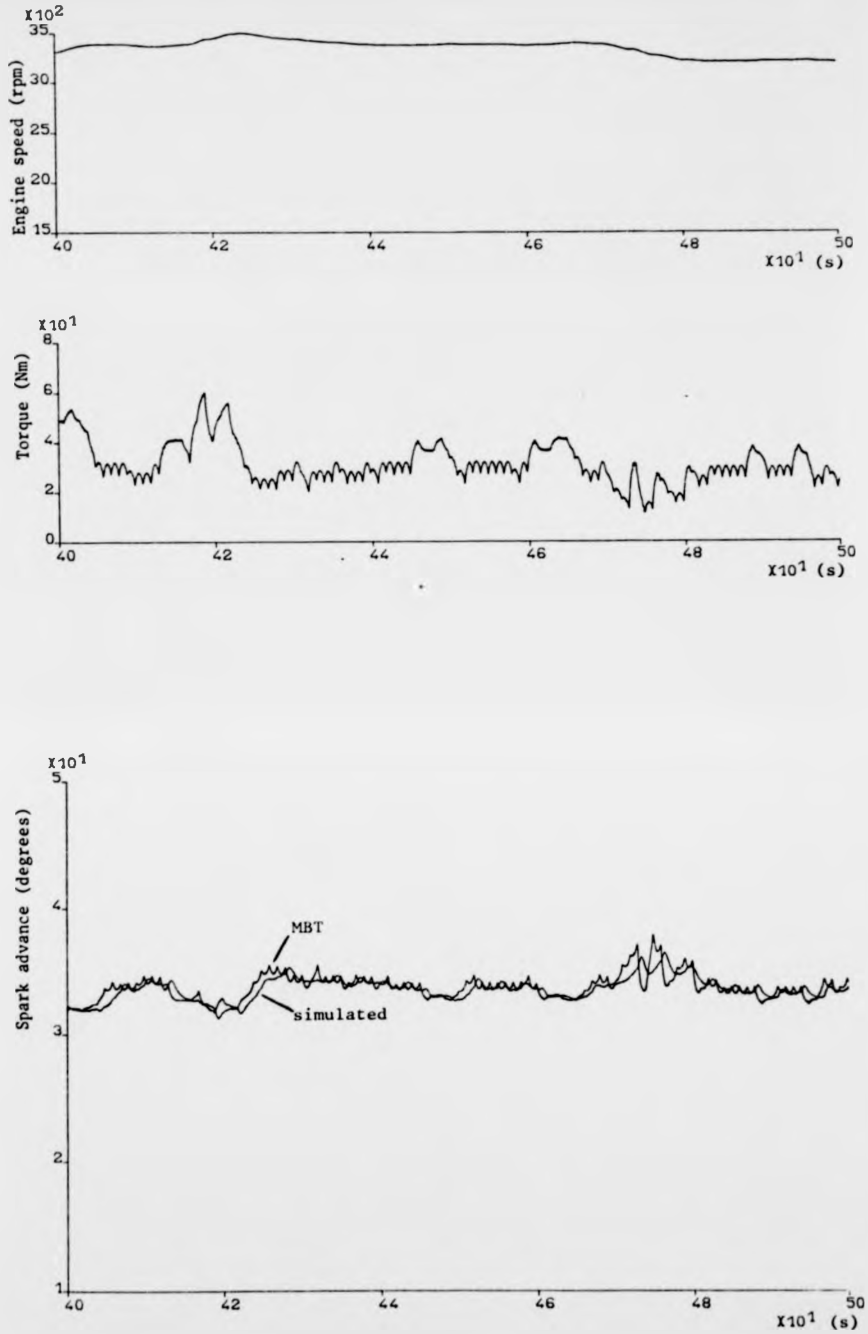


Figure 7.8 Controller performance on a portion of the EPA Highway schedule (perturbation amplitude = 3, gain K = 0.10)

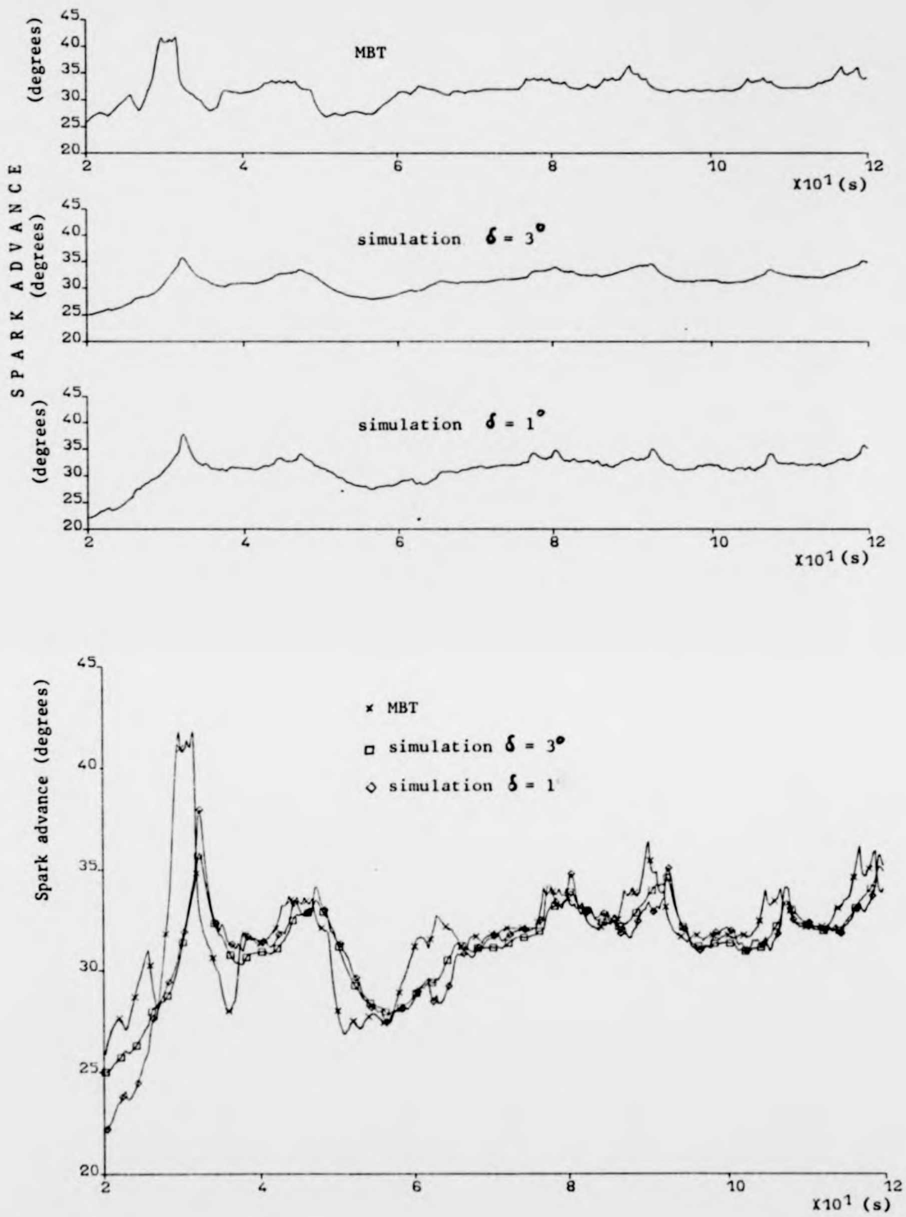
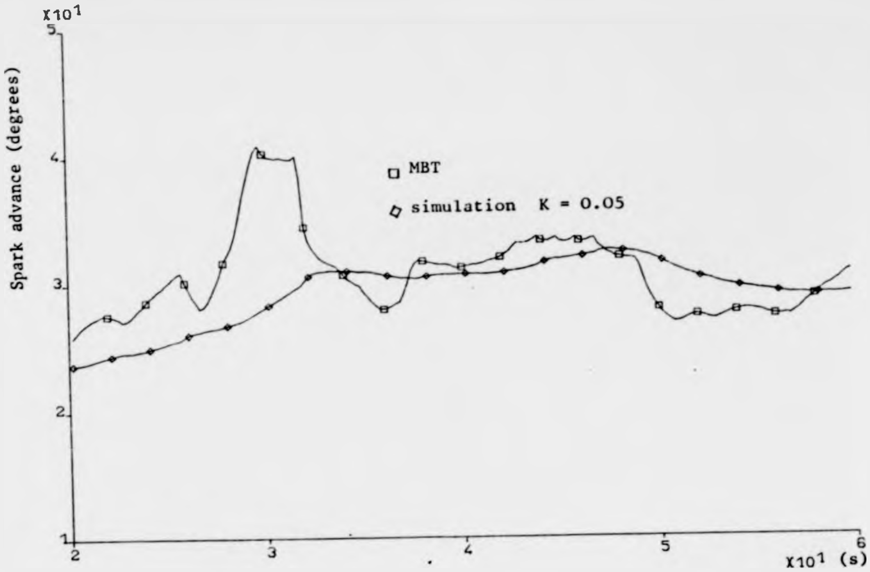


Figure 7.9 Influence of perturbation amplitude on controller performance



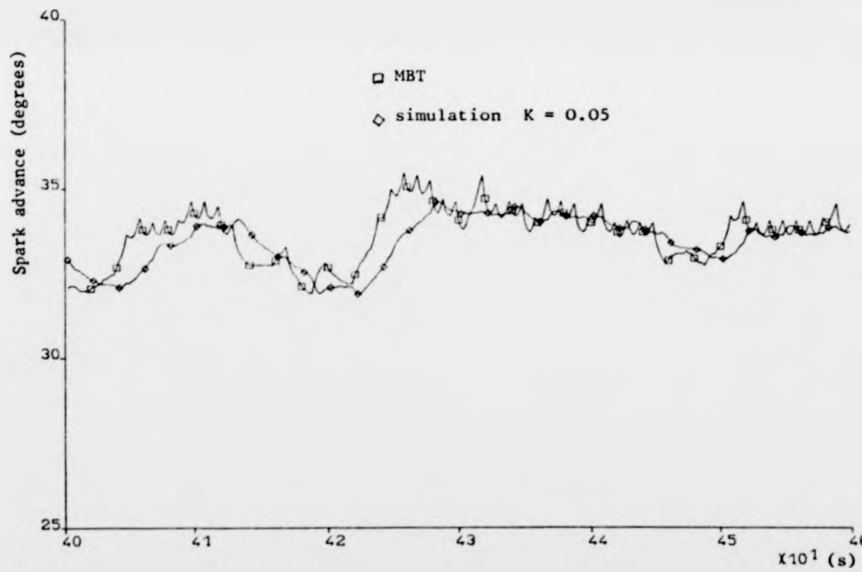
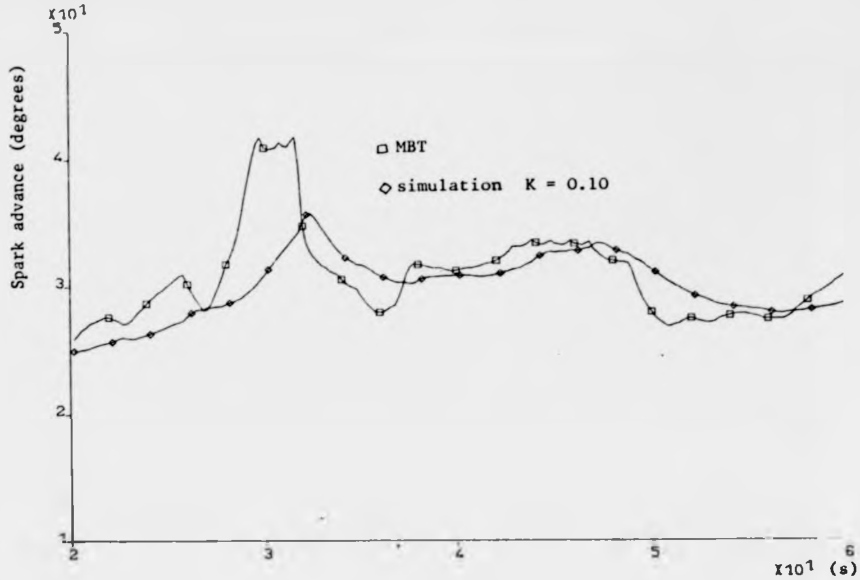


Figure 7.10 Two portions from EPA Highway schedule (gain K = 0.05)



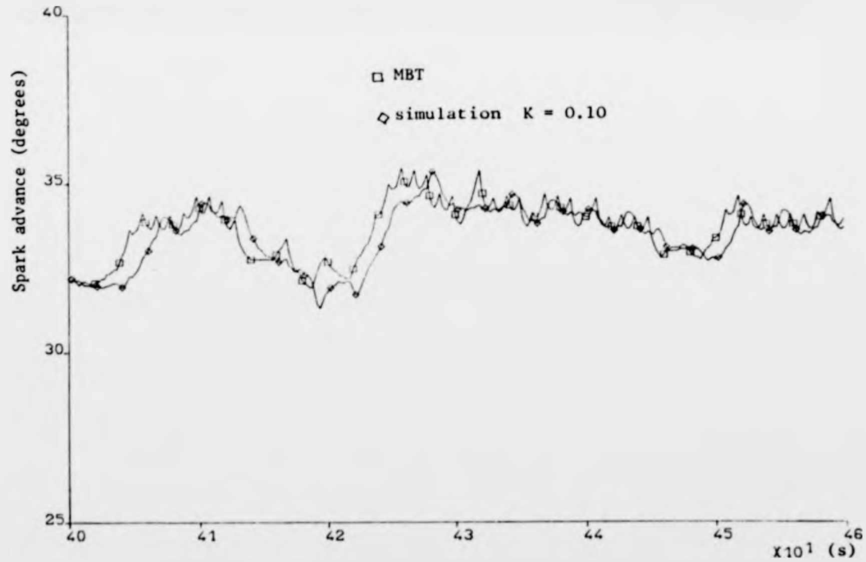
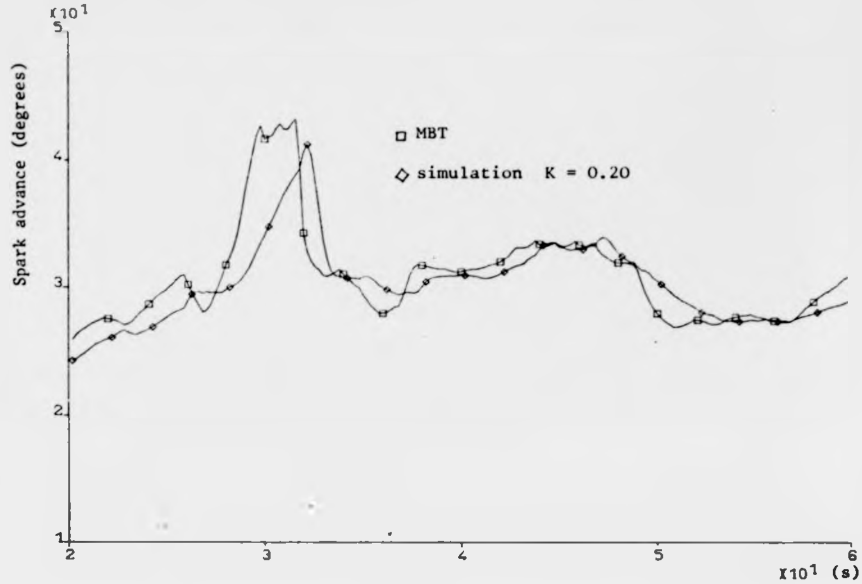


Figure 7.11 Two portions from EPA Highway schedule (gain K = 0.10)



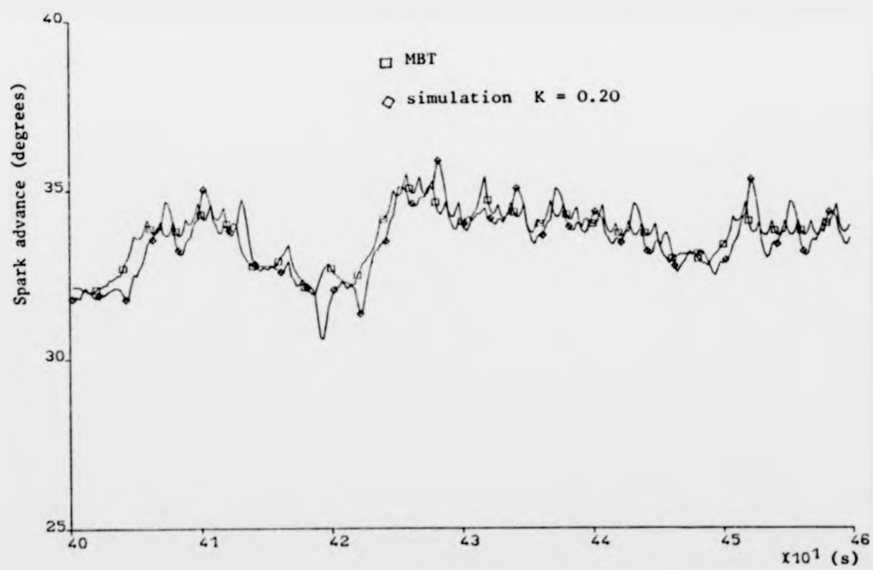


Figure 7.12 Two portions from EPA Highway schedule (gain K = 0.20)

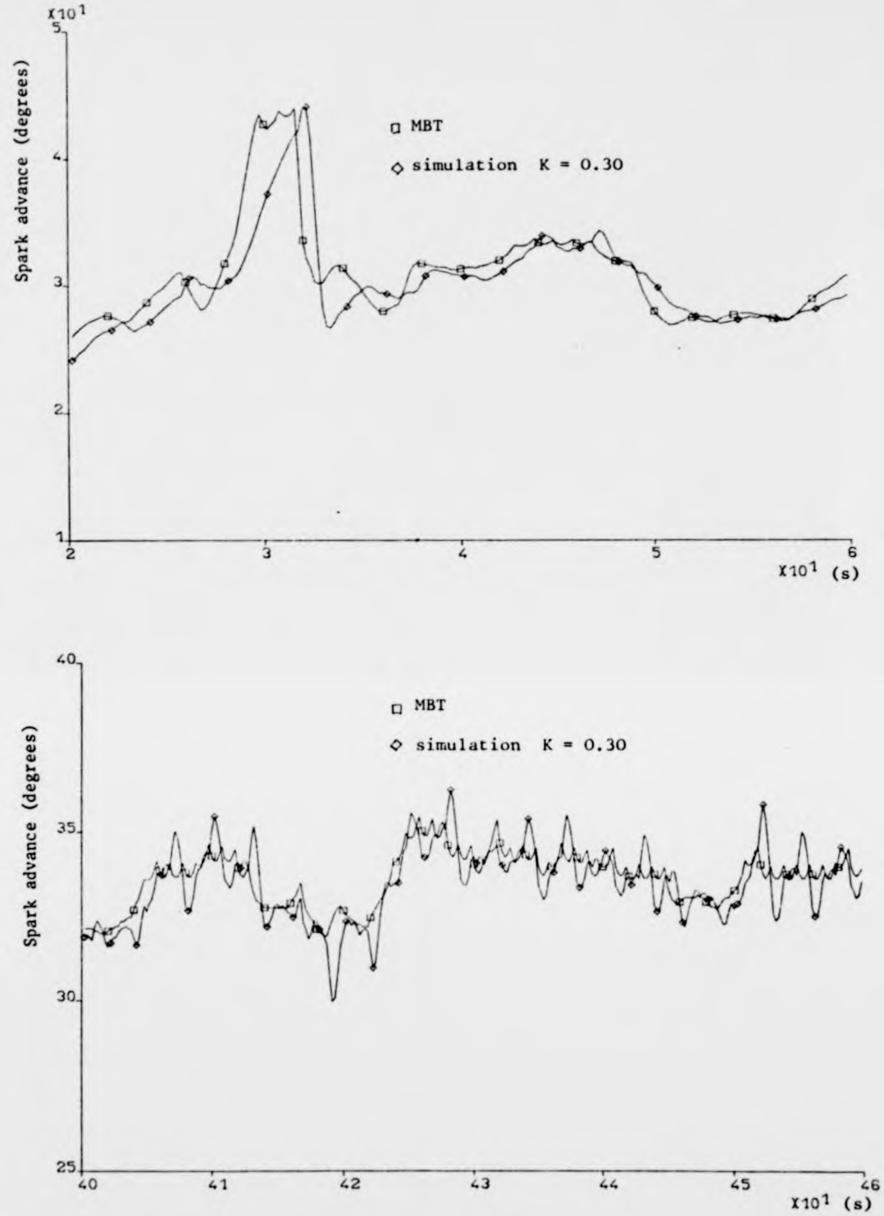


Figure 7.13 Two portions from EPA Highway schedule (gain K = 0.30)

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER WORK

8.1 REVIEW OF THE WORK.

With the increasing development of new vehicles with advanced powertrains, and the increasing prominence of energy and environmental considerations, sophisticated computer aided analysis and design tools are rapidly becoming essential. Computer aided engineering (CAE) in automotive applications has gradually extended its boundaries beyond geometrical modelling, finite element techniques, and drafting, into automotive engine management and driveline control. In this particular area of activity the CAE techniques proving most useful are static and dynamic systems modelling and simulation, and control systems synthesis and optimisation.

The research work forming the basis for this thesis, has sought in the main to concentrate on the engine control aspects of the fuel and emission performance of vehicles. An attempt, however, has been made here to ensure that engine control aspects are placed firmly in their correct position in the wider context of the control of the vehicle system as a whole; in fact it will be seen that the material in this thesis moves from the general considerations of factors influencing the energy efficiency of vehicles, through the considerations involved in the treatment of engine control problems, and onto application of simulation techniques in the study of specific engine control strategies. This progression from broad considerations towards consideration of detail, and the complementary procedure of successive refinement, are fundamental to the whole process of addressing engineering problems, involving for instance the modelling and simulation of a vehicle system.

A three-fold purpose to the work is evident from this dissertation, and may be summarised as follows:

i) to define the fuel economy problem with respect to legislative vehicle tests, and to assess the potential for reducing vehicle fuel consumption by the development of improved automotive systems.

ii) to determine the major requirements, and develop suitable models of road vehicles, and a methodology for addressing problems of emission constrained fuel economy optimisation; with particular regard to engine calibration/control approaches to the problem.

iii) to apply modelling techniques to the development of a dynamic automotive system simulation facility, suitable for addressing the type of problems defined; and to use the simulation facility in the study of certain engine control strategies for fuel economy optimisation.

These three areas of the work are discussed in the following sections.

8.1.1 Constrained Fuel Economy Optimisation and Vehicle Development Potential.

Chapter 2 provides a formulation of an emission and driveability constrained fuel economy optimisation problem. This is posed in terms of engine calibration, the solution of which involves the derivation of control laws to govern, typically, the engine ignition timing, air-fuel ratio and exhaust gas recirculation. In practice the problem may be considerably more complex owing to additional control variables such as gear ratio, or additional constraints such as those brought about by actuator hardware limitations. The formulation presented is adequate, however, for the scope of study encompassed by this work; and it is the type of engine control problem most frequently addressed.

Several approaches to this engine control problem are discussed, and almost all of these are based solely on steady-state engine performance data from warmed-up engines. A major limitation of these procedures is underlined in chapter 2: it is that they produce controls that commonly require radical modifications simply because relevant dynamic vehicle system characteristics have not, and usually cannot, be incorporated into the procedure. While these procedures do provide a useful first estimate of a feasible solution, it is desirable to have methods which can accommodate the dynamics of the vehicle powertrain and of the fuelling and emission production processes, in order to develop engine control laws which are more likely to meet the constraints on emissions and 'driveability', without subsequent modifications external to the optimisation procedure.

Apart from conventional open-loop feedforward controls, chapter 2 refers also to feedback air-fuel ratio controls which are less widely used; and to adaptive controllers, which despite certain obvious attractions, have had difficulty gaining acceptance on a wide scale for automotive engine control.

The complexities of producing a performance metric for comparing the fuel usage of different vehicles, is underlined in the first part of chapter 3. Both fuel and emissions performance are intimately related to vehicle duty and driver behaviour; the essence of the problem is that in removing sources of variability in order to produce a precise and repeatable test procedure, the test conditions and results are likely to diverge markedly from the reality of everyday driving.

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The performance metric chosen is often a driving schedule (or set of schedules) and certain steady vehicle speed conditions. It is not clear that the schedules based on examples of actual vehicle usage, such as the U.S. EPA schedules, are any more representative of fuel and emissions performance under every day conditions, than the somewhat artificial driving schedules used in Japan and Europe. Neither is there sufficient evidence to suggest that the emissions constraints imposed by various legislative bodies worldwide, have any direct relationship to the contribution made by automobiles to atmospheric pollution; the harmful effects of individual pollutants also have not been fully quantified.

It is the author's view that a substantial amount of work is still required in order to arrive at reasonable metrics for comparative vehicle performance assessment in terms of fuel economy, and as a basis for emissions legislation: this work must entail sufficient attention to the major aspects of vehicle dynamics, driver behaviour, road conditions and topography, if the results are to be at all useful.

Development Potential

The second part of chapter 3 concentrated on the potential development areas of automotive systems which can have significant effect on the fuel economy of the vehicle. Better energy management must be seen both as a long term and a short term goal: in the short term improvements are often made to rectify deficiencies of existing designs in order to meet recent legislation, while longer term development is more likely to be directed towards larger improvements and more permanent solutions to the problems of legislation and marketing forces.

Each element of the automotive system can be optimised with regard to the overall energy management of the system; but many of these elements cannot be optimised in isolation owing to significant interaction with other system elements, and to their effect on other constraints such as those associated with accommodation and driver comfort.

For convenience the automotive powertrain is considered as three major interacting subsystems: vehicle, transmission, and power plant - the effect of the driver, though significant, is largely uncontrollable and is not considered here.

The most significant areas for optimisation on the vehicle are the aerodynamic qualities of the body, and the vehicle mass; but as usual, compromises need to be made to meet requirements of accommodation, ride quality strength, noise and vibration, for example. Vehicle weight, aerodynamic and rolling resistance characteristics have a marked effect on vehicle performance; and if these parameters are optimised then an additional gain in fuel economy, for the same performance level, may be possible by matching a smaller power unit and transmission to the vehicle.

As far as the development potential of engines for saloon cars is concerned, it appears that there are only two reasonable alternatives to the normal spark ignition petrol engine: the gas turbine and the diesel engine. The gas turbine has the potential for better fuel efficiency than the petrol engine; but acceptable materials to cope with the very high inlet turbine temperatures, and a solution to the problem of obtaining fast dynamic response, need to be found.

The diesel engine currently has a fuel economy advantage over the petrol engine, but the latter has a cost, performance, noise and weight advantage, in this application. In addition the emissions legislation is a severe challenge to the current diesel engine technology.

The development potential of conventional petrol engines is considerable in the areas of air-fuel mixture preparation, combustion chamber design, engine controls, and exhaust after-treatment. These developments must go hand-in-hand with vehicle improvements and transmission design, as changes to vehicle mass and aerodynamics, and changes to overall gear ratio or transmission design have significant effect on the requirements of engine size and torque characteristics.

The biggest significant improvement to fuel economy with regard to transmissions, is likely to result from the use of well controlled continuously variable transmissions (CVT); there is likely to be an accompanying cost and weight reduction which will have some contribution to the fuel performance. The greater use of 4 or 5 speed automatic transmissions, with lock-up torque converters, will show significant improvements over conventional automatics; and the introduction of 'manual' gearboxes with up to 10 ratios has considerable potential for fuel saving, although these would probably require computer management of engine throttle, clutch and gear selection, if the potential was to be achieved. Both these systems would inevitably suffer from a disadvantage brought about by the cost of additional materials and complexity. In recent years the trend towards higher overall gear ratio has provided improvements in vehicle fuel economy (by more efficient engine operation); this trend is likely to continue and will inevitably affect engine design as there is a requirement for higher torque at lower engine speeds.

A major area which has an indirect, but potentially large, effect on fuel consumption, is the development of new and improved sensors for automotive use. The availability of suitable sensors is a key factor limiting the development and introduction of advanced techniques for managing vehicle functions such as engine and transmission control.

8.1.2 Automotive Systems Modelling.

Chapter 4 provides a methodical treatment of the modelling of automotive systems, with particular regard to engine control approaches to emission constrained fuel economy problems.

It is emphasised throughout the treatment of the modelling process that the scope of any simulation tasks to be performed, must be defined as a prerequisite. This information is vital if the system is to be efficiently described; avoiding unnecessary complexity or oversimplification of individual subsystems. This is perhaps where the first problem arises as it is not always obvious what level of model complexity is appropriate, for example: an engine model including a representation of flame propagation, is clearly overly complex if one merely wishes to look at the effect of gear ratio on vehicle acceleration; however, it may not be as clear if the model is to be used in simulations to assess an engine control algorithm incorporating adaptive behaviour to engine knock. Often such problems are resolved using engineering insight, (which usually can be considered as a priori knowledge), or by recourse to physical principles, experimental data, or a combination of all three.

Another important aspect of the methodology is the conceptual decomposition of the automotive system into a number of simpler interconnected subsystems. Usually it is best to subdivide the system on natural functional boundaries, recognising at the same time the physical interactions that occur between the subsystems. In chapter 4 the subsystems are chosen to be the vehicle, engine, transmission, controller, and driver; however, for simulation work involving very detailed modelling of the engine, for instance, it may be desirable to make further subdivisions of that particular 'module' into, say, carburettor, induction manifold, combustion chamber, etc.

The vehicle is modelled according to the well known 'road load' equation. This is usually perfectly adequate for engine control problems of fuel usage optimisation, but if optimisation of vehicle components was of concern then the model would have to include such parameters. The transmission in this context is considered, for convenience, to comprise all the rotating parts between the engine flywheel and the tyre-road interface, including the clutch and the tyres themselves. For this work a non-compliant model was chosen; however if driveability in its widest sense was a key issue in the simulation, then driveline dynamics would need representing. If, for example, studies resulted in 'total driveability' being defined, and formulated in a suitable way to include it as a constraint in a fuel minimisation problem, then a non-compliant model would clearly be of very limited use. The major sources of compliance in the powertrain typically are the tyres, engine and transmission mountings, clutch centre and driveshafts; each compliant element could be represented separately in terms of the stiffness and damping associated with it, or lumped with other elements, according to the role that the simulation is intended to fulfil.

Although the transmission model in chapter 4 incorporates gear efficiencies explicitly, there is a strong case in favour of representing losses in terms of torque, as indicated in the text. The concept of efficiency has proved popular as a normalised measure for comparative assessment of plant under steady state operation. It does, however, prove awkward to use in dynamic simulation because of its behaviour under certain conditions approaching zero load; and with the increasing use of CVT transmissions in advanced powertrains, some of which have a geared neutral capability, the problems are best avoided by explicit use of torque or power loss in favour of efficiency.

The modelling of the engine is obviously of considerable importance to simulation studies involving engine calibration and control. Many algorithms are available for producing engine control laws (calibration), based on steady-state engine data, but it is well known that calibrations produced by these methods require substantial manual tuning to give the designed performance on actual vehicles. This is attributed in the main to the absence of those engine and powertrain dynamics, which influence emission production, fuel consumption and driveability.

While engines are often routinely 'mapped' under steady state conditions, dynamic emissions and fuel flow data is not readily available. The author has attempted to identify and model engine dynamics that have significant effect on fuel and emissions flow, in order to integrate them into an engine model having steady state data as its basis. These dynamics largely involve actuator, induction, and temperature phenomena, which have a major effect during the warm-up phase of engine operation - a phase for which data is difficult to obtain and, at present, largely unavailable.

The author has found that the widely used techniques of multiple linear regression analysis are an excellent means of modelling steady state engine data, but great care is needed in their use. The main problem lies in the ease to which the techniques lend themselves to automation, inviting the engineer to accept the resultant model, simply on the basis of a favourable basic model statistic produced by the process. This danger is particularly prevalent in circumstances where the regressors used in the model are highly correlated, or have no real physical interpretation - this is often the situation with the modelling of engine map data.

The importance of interaction in the regression process, and the importance of a knowledge of the process itself and the physical behaviour of the engine, must not be under emphasised if a suitable model is to result. Perhaps the most important interaction in the process, is the validation of the model (or set of models) against the original data using graphical analysis techniques. The author has frequently found during this study and in subsequent work, that a model based entirely on multiple linear regression analysis, may be insufficient in some areas of engine operation. This has led to the need to augment the regression based model by imposing additional constraints on the function, or by modelling smaller portions of the data separately.

The controller module is considered as hierarchically superior to the vehicle, transmission, and engine. More than any other module, the controller is of arbitrary complexity, depending upon its role in the specific system. In this application its role is confined to the control of the ignition timing and air-fuel ratio of the engine, but in an advanced vehicle it may encompass the complex tasks involved in interpreting the driver's intentions and controlling the transmission ratio, clutch, engine throttle and braking system.

The driver is considered as residing at the highest level in the system hierarchy, and the function of the driver in this application was to track any given driving schedule. Chapter 4 considers this function in terms of four tasks: gear selection, throttle setting, clutch control and brake setting. The individual characteristics of the driver and his interaction with the vehicle, are considered to be embodied in the driving schedule; such that the driver model should simply ensure that the schedule is tracked to acceptable accuracy. The exception to this is the treatment of gear selection, which is not considered to be (unrealistically) bound to gear change points which may be specified by the driving schedule.

The gear change algorithm presented, is an important feature of the driver module and copes well with 'mild' and 'aggressive' driving schedules. The driver model does not exhibit a learning behaviour, but is intended to represent a driver who has become fully acquainted with the performance of the vehicle - this is not unrealistic in light of the fact that a driver, when performing a government fuel economy test on a new vehicle, will rehearse the exercise prior to the actual test.

The driver model as represented in chapter 4 has performed exceptionally well in simulations involving arbitrary driving schedules; but it must be borne in mind that there are many subtleties of clutch and engine speed control, which while mastered by actual drivers, are not represented here. It follows that if driveline compliances were present in the model, further consideration would be needed to the modelling of these aspects of driver function.

8.1.3 The Development of an Automotive System Simulation.

The development of the Continuous Automotive System Simulation (CASS) was a major portion of the work reported here. This included the development and coding of the models, and the design and development of the simulation facility into which they were integrated.

In the modelling phase it was necessary to build software tools to aid in the development of the engine models. These enabled a sequence of regression models to be developed semi-automatically, with interaction at key points in the process in order to control and monitor the accuracy of the resultant models. Even with such computational and graphical analysis tools, the process from the raw data to the final models of the engine variables, was found to be lengthy; and the large quantities of data, to be handled during the process, required careful maintenance.

An attempt was made to produce a modular simulation facility, primarily by the hierarchical representation of the automotive system, and the implementation of the program code using subroutines; and secondly by the separation of model parameters into files to be read by the simulation program prior to beginning a run. The development of the program as a set of subroutine modules is intended to permit incremental development and testing of the facility, and to allow subroutines representing different parts of the automotive system to be readily replaced if, say, a different vehicle was to be simulated. The separation of model parameters from the model definition permits these parameters to be altered and a new simulation to be performed, without the need to recompile the simulation facility. The comprehensive output from the model provides a brief summary of the vehicle parameters, and fuel and emissions performance, as well as providing the trajectory of a large set of variables, in a compact form suitable for data analysis using the Interactive Data Analysis (IDA) package developed at the University of Warwick.

CASS proved easy to operate, with the user responding to prompts for the names of the files containing the engine-vehicle data, the driving schedule, and the control coefficients. Parameters could be changed directly in the files using a text editor, and new engine calibrations incorporated in a similar manner.

8.1.4 Application: Study of an Engine Calibration Technique.

Dynamic simulation is a valuable tool for use early in the study and assessment of strategies for the control of automotive systems. A study of one particular algorithm intended for on-line use for engine calibration, is reported in chapter 6. This study was aided by the simulation facility C.A.S.S. developed by the author, and by the powerful data analysis and computer graphics facilities provided by the Interactive Data Analysis (IDA) package, developed at the University of Warwick.

The familiar problem of the constrained minimisation of fuel consumption, is addressed using a first-order gradient algorithm. The approach to the formulation of the optimisation procedure is more direct than that reported by Dohner (8.1), but yields a similar algorithm. The method relies on a facility for continuously measuring fuel flow, emissions, and other engine variables as the vehicle is driven over the prescribed driving schedule. By replicating driving schedule runs the local sensitivity of the fuel and emissions to the engine controls (air-fuel ratio, spark advance) may be determined, and an 'improved' control derived on the basis of a step taken in the downhill direction with respect to the cost function.

The algorithm is designed such that only few function evaluations are required at each iteration; as function evaluations are costly both in time and facilities, in the intended application of the method, i.e. to on-line engine calibration. The power of simulation techniques in assessing the performance of such a strategy, lies in the greater control over the experimental conditions and repeatability, the ease and speed with which experiments can be performed, and the ease with which control modifications can be implemented and tested.

The study concentrated on aspects related to the implementation and performance of the calibration procedure. In particular the effect of varying emissions constraints on the performance, and the influence of certain numerical procedures used in generating a sequence of improved controls.

The actual emissions constraints used in the work, and the cumulative fuel and emissions results for individual simulation runs, should be interpreted qualitatively, as it must be stressed that a specific vehicle is not being simulated. It is also desirable to make a full assessment of a control procedure such as this on the basis of simulation results using models of a number of alternative vehicles; corroborating the findings with subsequent tests on actual vehicles, if the procedure shows sufficient potential.

Generally a large reduction in the cost function was obtained on the first iteration for tests involving a wide range of emissions constraints. In many tests where the emission constraints were realisable, the emissions were satisfied on the first iteration; however, during subsequent iterations it was found that sometimes one or other of the emissions would 'creep' back above the level of the constraint, with only a negligible improvement in fuel economy. The application of a severe, and unrealisable, constraint again led to the major improvement being made on the first and second iterations, with often very little improvement or even degradation occurring, on subsequent iterations.

When constraints were used, that were more easily met by the system, then improvements in fuel consumption were made, but the convergence rate was disappointing. Convergence itself could be detected by a minimal decrease in fuel consumption with emissions constraints satisfied. Conversely, severe constraints resulted in a failure to reduce emissions to satisfy the constraint, or more usually, by the failure of the algorithm to produce a significant variation in the control.

Although the algorithm was designed to minimise the number of function evaluations at each iteration owing to their inherent cost, there are a number of initial runs required to obtain a suitable weighting matrix. Some of these runs (or function evaluations) establish a scaling to account for the relative sensitivity of the cost function to the control (calibration) variables, and other runs form a one-dimensional search to determine a suitable step size.

It is perhaps not surprising that the algorithm produces a good first step, when it is considered that the weighting matrix contains information from the locality of the reference run, including an approximate line search to determine a suitable step size. On subsequent iterations the weighting matrix is not re-evaluated, and no line search is performed on account of the high cost, thus reducing the effectiveness of the iterative step.

If the number of function evaluations (or runs) was not as important as it is in an on-line procedure, then convergence could be considerably improved by updating the weighting matrix and performing a line search at each iteration, to create a quasi-Newton algorithm. Under these circumstances (with a relaxed constraint on the number of function evaluations) a number of better algorithms would have possible application to the problem.

In the present context, the constant weighting matrix and lack of line search, are likely to be severe limitations to the performance of the algorithm. It is also true that the gradient direction is not always the 'best' direction in which to proceed, by reason of the fact that the gradient is a local property, and that steepest descent algorithms can sometimes be obstructed by certain non-linearities. It is intuitive that if the weighting matrix retains information from previous iterative steps, then it is likely to be of greater value in defining the next step, than a matrix which has information only about the system in the vicinity of the reference run. A useful improvement to the algorithm may be expected, therefore, if a suitable updating scheme, based on the gradient information for previous iterations, were implemented. It is also possible with this algorithm to compare predicted changes in e and J (from equations 6.9, 6.11) with actual changes - too small a discrepancy implies that the step size may be usefully increased and, conversely, too large a discrepancy indicates that a reduction in step size is needed.

8.1.5 Application: Study of a Simple Adaptive Engine Control Scheme.

A second application of C.A.S.S. to engine control is reported in chapter 7. This is an elementary adaptive ignition timing controller, of the extremum-seeking type. As such the engine control problem could not be posed in the familiar context of emission constrained fuel minimisation; instead an unconstrained fuel minimisation task is implicit in the control scheme. The objective of the controller is simply to adjust the ignition timing to a value which achieves maximum output torque, hence optimum engine efficiency with respect to spark advance, and optimum specific fuel consumption.

The ignition timing which achieves this objective is termed the minimum best torque (MBT) spark advance, and it is intimately related to the engine operating conditions. Although the operating conditions of speed and load are fundamental to MBT timing, other stochastic and time dependent variables have significant effect, such as air temperature, atmospheric pressure, and fuel characteristics. This variability of plant and operating conditions and the ability of adaptive controls to respond to such unsensed variables, is the chief attraction of adaptive schemes.

The type of perturbation-driven controller simulated as part of this study is not new, but the decision to incorporate such a controller in the automotive simulation facility, was taken in order to

a) make a basic evaluation of the controller's performance under simulated driving schedule conditions.

b) demonstrate the wide applicability of simulation techniques in control system synthesis and performance assessment.

c) determine any additional considerations for modelling and simulation of this type of controller and, in particular, to determine the suitability of CASS in this application.

The controller operated by superimposing a sinusoidal or rectangular perturbation on the nominal ignition timing, and correlating the behaviour of the engine 'output' with this perturbation. This provided a measure of the gradient of the relevant engine characteristic in the locality of the operating point, and enabled nominal ignition timing to be processed in a direction tending to maximise engine output power.

The first difficulty was found to be the selection of a suitable engine variable to be measured as a basis for the controller. Engine speed was initially chasen, as flywheel acceleration is dependant on engine output torque and it is the easiest candidate to measure in pratice. Its unsuitability was evident from initial attempts to perform simulations using the simple controller; and this can be attributed both to the non-compliant representation of the vehicle system, and to the fact that no torque converter or fluid coupling is present between engine and transmission. The subsequent choice of engine torque as the output variable to be maximised was excellent from the simulation standpoint, but it is a greater problem to find a suitable torque sensor for practical applications. However, flywheel speed may also prove difficult to use (unless considerable compliance is present between engine and transmission) owing to the high attenuation of the frequencies of interest, by the major inertias in the powertrain.

Predictably the performance of the controller was good at near steady state engine conditions, but poor under transient operation which was found to apply to much of ECE-15, EPA Urban, and EPA Highway schedules The worst behaviour was noted during gear changes, when the simulated ignition timing bore very little relationship to the calculated MBT timing.

Despite the unsuitability of this controller for normal road vehicles, it is potentially useful for application to engines which have reasonably steady operating duty, and possibly need to operate under widely varying ambient conditions. Other potential applications lie in the field of engine testing, where such a controller may be used to rapidly determine MBT ignition timing for a wide range of operating conditions for the engine under test; or may be used to maintain the engine at the optimum timing while investigating the effect of parameter tuning or component modifications elsewhere on the engine.

It was found that the filter characteristics needed to be chosen fairly carefully if the best controller performance was to be obtained. The perturbation frequency did not, however, seem to be critical on the system represented in CASS.

8.2 RECOMMENDATIONS FOR FURTHER WORK.

Although a broad view of vehicle development potential with particular regard to fuel economy is presented initially, the main portion of this study has of necessity concentrated on some aspects of a specific area of vehicle development. These aspects relate to the development of engine controls, typically addressing problems of emission constrained fuel economy.

The processes involved in the development of automotive system models, the development of the simulation facility, and the application of simulation to specific problems, have served to clarify many general and specific requirements of automotive systems engineers. Advanced automotive vehicles have an inherent complexity which necessitates the use of sophisticated computer based tools in order to design control strategies which optimise the overall system performance. A minimum requirement is a facility for dynamic nonlinear simulation for exercising mathematical models of the automotive powertrain of interest. There is also a need for powerful data management and processing tools, as well as computer aided control systems design facilities (CACSD). Ideally these tools would be integrated into a computer aided engineering environment, tailored specifically to the requirements of automotive systems engineers.

In general the processes involved in developing models of vehicle systems cannot be automated, as many tasks are iterative and require human interaction. The modelling of vehicle subsystems is usually a combination of physical principles, empirical and design data, and a priori knowledge; but often the largest tasks are associated with the design of experiments, the collection of data, and the analysis and modelling of that data. Methods and procedures for obtaining steady state data from vehicle systems are widely practiced, but there is an urgent need for effective methods of identifying systems in terms of the dynamics of the engine and powertrain, in order to address the control problems via nonlinear simulation techniques.

While the work reported here did not involve the author in a major programme of vehicle instrumentation and data collection, there were extensive data processing needs to be met once the data had been obtained. The CAE environment referred to above would need versatile data entry facilities for (low volume) keyboard entry, (high volume) magnetic tape, and for data access via computer networks. In view of the nature of vehicle data a management system capable of handling large quantities of data is desirable: it would need to cope with data as diverse as component specifications, engine maps, and time series records; and it would ideally be able to access data according to the relationships which exist between the items in the database.

During the modelling and analysis stage of this study many computer programs were developed for manipulating, filtering, regression analysis and other processing tasks to be performed on selected data. While such tasks are frequently highly interactive, they represent processes which would be repeated whenever empirical data from other vehicles or from simulation were to be analysed or modelled. It therefore follows that an interactive automotive data analysis and modelling facility, which is expandable to incorporate new processes, would be an invaluable tool for automotive systems engineers engaged in modelling and simulation.

The chief deficiencies of the analysis and modelling process that formed a significant part of this study, lay in the custom-built nature of the software tools themselves, and in the means of managing the very large quantities of data. The development of the data management system referred to above and its integration with an automotive data processing facility, would greatly improve the speed and efficiency of this vital stage in control system development; obviating the need for the user to create and keep track of data structures, and to develop or adapt programs for his specific processing requirements. A vital element of the analysis facility in the proposed CAE environment, is that of versatile interactive graphical display facilities - this was found to be by far the most important feature offered by the Interactive Data Analysis (IDA) facility developed at the University of Warwick.

It has been stated that a minimum requirement for addressing the control problems of automobiles is the availability of a facility for simulating the dynamics of nonlinear systems. CASS was designed with the intention of providing a fair degree of versatility; in that an attempt is made to represent the automotive system as a set of smaller subsystems, the coding employed a corresponding set of subroutine subprograms, and the model and driving schedule parameters could be altered without re-compiling the program. In practice it was found that CASS was far from ideal, even for the applications to which it was put in this study: basically it was too limited in its flexibility, both from the point of view of the simulation facilities if offered, and the ease with which system (model) changes could be incorporated.

Experience with CASS has enabled a number of desirable features to be defined for the future development of tools for automotive system modelling, simulation, and control. These features relate to the simulation facility itself, and to the proposed environment in which it would reside.

With regard to the simulation facility:

a) the formulation of the automotive system simulation model should be independant of the simulation facility which is used to exercise it. General purpose scientific languages such as Fortran are awkward to use for model formulation, and a more convenient high level language, specifically for dynamic simulation, should be adopted.

b) interactive means of exercising simulation models should be possible, and it is essential to be able to alter model parameters and continue a simulation without the need to re-generate an executable program.

c) automotive systems involve a wide range of dynamics - it is thus essential to have effective means of handling stiff dynamic systems. It is also desirable to be able to specify the parameters which control the simulation process, conveniently and rapidly; and also to be able to monitor and control the output of simulation results in an interactive manner.

Two considerations with regard to the CAE environment have already been mentioned above: an extensive and secure database management system, and a versatile data analysis facility. Additionally the environment should support a modular approach to the development of automotive simulation models, recognising natural and convenient subdivisions of the total vehicle system into an arbitrary number of interacting subsystems.

d) allow for the incremental development of existing facilities and the addition of new facilities.

e) incorporate a versatile control system design suite.

f) incorporate an interface to a facility for producing high integrity software for implementing digital control algorithms on a wide range of target microprocessor-based controllers.

Above all the whole environment should be reliable, secure, and provide a man-machine interface which is 'user-friendly' - having consistency and predictability in its interaction with the user, an extensive help system, and a capability of being used effectively by engineers with arbitrary familiarity with computers or with the particular host computer.

The availability of the whole range of CAE facilities referred to above, whether or not they are integrated into an 'environment', would enable a wide range of automotive control system problems to be addressed in an effective manner; although productivity and efficiency would undoubtedly be enhanced by the integration of the package, as the problems involved in managing the large amounts of data are significant , and it often proves difficult to interface utilities performing separate processing tasks. Strategies for calibration and engine control such as those studied here using CASS, could be studied more thoroughly within the proposed environment. However, calibration and powertrain control system design processes must undergo considerable development, before a stage is reached where controllers developed on the computer can be expected to perform well on the vehicle without substantial additional tuning.

One of the major hindrances to good controller development is the lack of adequate 'driveability' models. If driveability, which is to a large degree a subjective phenomena, could in some way be quantified, then it could be used as a constraint in the optimisation of engine and powertrain controllers; thus greatly reducing the need for tuning following installation of the controller. A fairly modest programme of work involving the installation of low cost instrumentation on a number of different vehicles would provide data which could be analysed to develop a useful set of driveability constraints.

The driveability issue is seen to be a limiting factor in all calibration methods used so far, including the on-line method studied here. This method does permit certain system dynamics to be handled, but as it does not use any system models, the whole procedure for obtaining an engine calibration must be repeated whenever the hardware is modified. Simulation techniques have the potential for rapidly generating control schedules for any given vehicle, if the modelling techniques are available for rapidly and accurately characterising the vehicle - considerable development is needed in this direction.

Clearly the potential for extremum-seeking controls such as the 'optimalizer' studied, is low in passenger car applications, with the possible exception of hybrid vehicles, where the engine may well be operated in a near steady state condition. However, the combination of feedforward control and adaptive principles has considerable appeal - the former providing the necessary speed of response, and the latter coping with the problems of drift, warmup, ageing, manufacturing tolerances, for example.

Finally, it is appropriate to end this thesis by referring to a subsequent programme of work in which the author has been engaged, and which has beed strongly influenced by the material and recommendations resulting from this study. During the course of the work the author has developed a 'user-friendly' computer aided engineering (CAE) environment known as 'Dynamic Automotive Powertrain Simulation' (DAPS, reference 8.2). It consists of a number of interactive utilities under the control of a run-time executive, essentially providing the requirements a-e above. It provides for the automated management of automotive subsystem modules and simulation results with a minimum of user effort, and gives access to the comprehensive simulation facilities within the Advanced Continuous Simulation Language ACSL (8.3).

DAPS has been used in applications involving the control of an advanced vehicle equipped with a continuously variable transmission (CVT), and the control of a heavy goods vehicle equipped with a discrete-ratio gearbox. The CAE environment and these applications are described in a number of papers (Appendix C and references 8.4-8.8).

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APPENDICES

APPENDIX A

LINEAR LEAST SQUARES REGRESSION

Given a system variable Y that can be represented by a linear combination of predictors X

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \ldots + \beta_k X_k + \varepsilon \qquad \text{(A.1)}$$

the problem can be defined as: determine the coefficients β that minimise some error ε.

An estimate $\hat{Y}$ of Y can be provided by

$$\hat{Y} = b_0 + b_1 X_1 + b_2 X_2 + \ldots + b_k X_k \qquad \text{(A.2)}$$

The least-squares solution attempts to choose $b_0, \ldots, b_k$ to minimise the sum of the square of the estimation error at each data point

$$Q = \sum_{i=1}^{n} (Y_i - \hat{Y}_i)^2 \qquad \text{(A.3)}$$

where Y_i is the ith measurement of a set of n measurements.

Single predictor case

The model of the data is

$$\hat{Y}_i = b_0 + b_1 X_1 \qquad \text{(A.4)}$$

where equation A.3 becomes

$$Q = \sum_{i=1}^{n} (Y_i - b_0 - b_1 X_i)^2$$

Equating the partial derivatives $\partial Q/\partial b_0$ and $\partial Q/\partial b_1$ to zero we obtain the normal equations

$$\begin{aligned} \sum Y_i &= nb_0 + b_1 \sum X_i \\ \sum X_i Y_i &= b_0 \sum X_i + b_1 \sum X_i^2 \end{aligned} \qquad (A.5)$$

which can be solved for b_0 and b_1.

In matrix notation for n observations equation A.2 becomes

$$\begin{bmatrix} \hat{Y}_1 \\ \hat{Y}_2 \\ \cdot \\ \cdot \\ \cdot \\ \hat{Y}_n \end{bmatrix} = \begin{bmatrix} 1 & X_1 \\ 1 & X_2 \\ \cdot & \cdot \\ \cdot & \cdot \\ \cdot & \cdot \\ 1 & X_n \end{bmatrix} \begin{bmatrix} b_0 \\ b_1 \end{bmatrix}$$

where X_i is the value of the predictor for the ith observation.

Equation A.3 can be written

$$Q = (Y - \hat{Y})^T (Y - \hat{Y})$$

and the normal equations as

$$W^T W B = W^T Y \qquad (A.6)$$

where $Y = [Y_1, Y_2, \ldots Y_n]^T$; $B = [b_0, b_1]^T$ and

$$W = \begin{bmatrix} 1 & X_1 \\ 1 & X_2 \\ \cdot & \cdot \\ \cdot & \cdot \\ \cdot & \cdot \\ 1 & X_n \end{bmatrix}$$

From A.6 we have the solution

$$B = [W^T W]^{-1} W^T Y \qquad (A.7)$$

requiring $W^T W$ to be non-singular.

Multivariate case

The response variable is often a linear combination of two or more predictors as in equation A.2. Fortunately equation A.7 still applies if we re-define W and B as

$$W = \begin{bmatrix} 1 & X_{11} & X_{12} \cdots & X_{1k} \\ 1 & X_{21} & X_{22} \cdots & X_{2k} \\ \vdots & \vdots & \vdots & \vdots \\ 1 & X_{n1} & X_{n2} & X_{nk} \end{bmatrix} ; \qquad B = \begin{bmatrix} b_0 \\ b_1 \\ \vdots \\ b_n \end{bmatrix}$$

The Coefficient of Multiple Determination

A measure of the success of a regression can be found in the coefficient of determination R^2, and by examination of the residuals $Y_i - \hat{Y}_i$. The coefficient of determination statistic is defined as

$$R^2 = \frac{\sum(\hat{Y}_i - \bar{Y})^2}{\sum(Y_i - \bar{Y})^2} = \frac{\text{variation in Y explained by regression}}{\text{total variation in Y}}$$

where $\bar{Y}$ is the mean of the set of n measurements of Y.

R^2 varies between zero and unity according to how well the regression explains the variation in the response variable Y.

The standard error of estimate is defined as

$$S_{y.x} = \frac{\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2}{n - (k+1)}$$

where k is the number of predictors, and the term

$$\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2$$

is known as the residual sum of squares, or the unexplained variation in Y.

It should be noted that

$$\sum(Y_i - \bar{Y})^2 = \sum(\hat{Y}_i - \bar{Y})^2 + \sum(Y_i - \hat{Y}_i)^2$$

total variation	explained variation	unexplained variation

It is, however, dangerous to assume that the parameter estimates are good simply on the basis of a high R statistic, when multicollinearity is present (strong correlation between regressors); this is of course the case when any polynomial model of the data is being used. Adding more regressors to the model will automatically increase the R^2 coefficient, regardless of whether the regressors bear any relation to the data being modelled. It is wise in any event to examine the whole profile of diagnostics available, in order to determine the significance of the individual parameter estimates b , with a view to including as few regressors as possible consistent with acceptable model performance; reference A.1 provides details of the use of F and t tests in this context.

Backward Elimination

Given a set of predictors $X_1,\ldots,X_k$ it is generally desirable to choose the smallest subset that will provide a satisfactory estimate of Y.

One method is known as backward elimination. A regression is performed initially with the full set of predictors, and a predictor is eliminated using a test of statistical significance (t-test or F-test). A regression is performed using the subset and the process of elimination is repeated. At each stage the R^2 coefficient and the estimation error is monitored until a point is reached where all the coefficients are significant.

It is important that after a regression has been performed, a plot of the residuals $Y_i - \hat{Y}_i$ is examined. If the residuals are heterogenous or there is an identifiable trend then it is likely that an important predictor has been omitted. The occurrence of out-lying data points requiring investigation will also be readily apparent.

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APPENDIX B

GENERAL SOLUTION OF THE STATE VECTOR DIFFERENTIAL EQUATION

Consider the state-space representation of a linear dynamic system

$$\dot{x}(t) = Ax(t) + Bu(t)$$
$$y(t) = Cx(t) + Du(t) \tag{B.1}$$

where
- $x(t)$ is the state vector
- $y(t)$ is the output vector
- $u(t)$ is the control or input vector
- A is the time independent coefficient matrix
- B is the driving matrix
- C is the output matrix
- D is the transmission matrix

Taking the Laplace transform, we have

$$sx(s) - x(0) = Ax(s) + Bu(s)$$

or

$$(sI - A)x(s) = x(0) + Bu(s) \tag{B.2}$$

The free motion of $x(t)$ (i.e. for $u(t) = [0]$)

$$(sI - A)x(s) = x(0)$$

which has the solution

$$x(t) = \exp[At] \cdot x(0)$$

The forced motion of $x(t)$, with $x(o) = [0]$ gives

$$[SI - A]\, x(s) = Bu(s) \tag{B.3}$$

for which it is desirable to have a solution of the same form as the free motion case

i.e. $$x(t) = \exp[At] \,.\, x_q(t) \tag{B.4}$$

where $x_q(t)$ is a vector of unknown functions.

Now

$$\frac{d[\exp[At].x_q(t)]}{dt} = A\exp[At].x_q(t) + \exp[At].\dot{x}_q(t)$$

which transforms to

$$s\,\mathcal{L}[\exp[At].x_q(t)] = A\,\mathcal{L}[\exp[At].x_q(t)] + \mathcal{L}[\exp[At].\dot{x}_q(t)]$$

or $$(sI - A)\,\mathcal{L}[\exp[At].x_q(t)] = \mathcal{L}[\exp[At].\dot{x}_q(t)]$$

Substituting B.4 into this gives

$$(sI - A)x(s) = \mathcal{L}[\exp[At].\dot{x}_q(t)]$$

Substituting the left hand side from B.3 and taking inverse transforms gives

$$Bu(t) = \exp[At].\dot{x}_q(t)$$

or

$$x_q(t) = \int_0^t \exp[A\tau].Bu(\tau).d\tau$$

Thus from B.4

$$x(t) = \exp[At] \int_0^{\tau} \exp[-A\tau].Bu(\tau).d\tau$$
$$= \int_0^{\tau} \exp[A(t-\tau)].Bu(\tau).d\tau$$

a convolution integral.

As we are considering a linear dynamic system, the principle of superposition holds; thus we can add the forced and unforced terms to give the total solution to B.1

$$x(t) = \Phi(t)x(0) + \int_0^{t} \Phi(t-\tau).Bu(\tau).d\tau \qquad (B.5)$$

where

$$\Phi(t) = \exp[At] = \mathcal{L}^{-1}[(sI - A)^{-1}]$$

the state-transition matrix, which may be determined by the series expansion

$$\Phi(t) = \exp[At] = I + At + \frac{(At)^2}{2!} + \ldots + \frac{(At)^r}{r!} + \ldots \qquad (B.6)$$

or by determining the inverse transform of each element of the inverse of the matrix (sI - A), or by using spectral resolution of matrix A (eigenvalue analysis).

The solution B.5 could be found directly from equation B.2

$$x(s) = (sI - A)^{-1}x(0) + (sI - A)^{-1}Bu(s) \qquad \text{(rearranged)}$$

by using the principle that 'multiplication in the frequency domain is equivalent to convolution in the time domain'.

Discrete Time Model

For implementation of the state space representation of a linear system on a computer, a discrete time model must be used. The signals are then considered in sampled form, being available only at discrete intervals.

Choosing the sample interval T (to give a good compromise between computing time and accuracy), it is convenient to approximate $u(\tau)$ (in equation B.5) as a constant between successive samples. That is, if

$$t_k = kT \;, \quad k \text{ (integer)} \geq 0$$

then

$$u(\tau) = u_k \;, \quad t_k < t < t_{k+1}$$

This approximation gives $u(\tau)$ as a series of steps, and in particular B.5 may be written

$$\begin{aligned} x(t_{k+1}) &= \exp[AT].x(t_k) + \int_0^T \exp[A(T-\tau)].Bu_k.d\tau \\ &= \exp[AT].x(t_k) + [-\exp[A(T-\tau)]]_0^T A^{-1}Bu_k \\ &= \exp[AT].x(t_k) + (\exp[AT] - I)A^{-1}Bu_k \end{aligned}$$

or

$$x(t_{k+1}) = \Phi(T)x(t_k) + (\Phi(T) - I)A^{-1}Bu_k$$

where

$$\Phi(T) = \exp[AT]$$

This may be written

$$x(t_{k+1}) = A_1 x(t_k) + B_1 u_k \qquad (B.7)$$

where

$$A_1 = \Phi(T)\ , \quad B_1 = (\Phi(T) - I)A^{-1}B$$

A_1 may be calculated from the series expansion

$$A_1 = \exp[AT] = I + AT + \frac{(AT)^2}{2!} + \ldots + \frac{(AT)^r}{r!} + \ldots \qquad \text{(B.8)}$$

$$B_1 = [\exp[AT] - I]A^{-1}B$$

$$= T[I + \frac{AT}{2!} + \frac{(AT)^2}{3!} + \ldots + \frac{(AT)^r}{(r+1)!} + \ldots]B \qquad \text{(B.9)}$$

Thus the inversion of A is avoided. The expansions are truncated where necessary to give the desired accuracy.

APPENDIX C

COMPUTER AIDED MODELLING AND SIMULATION OF AUTOMOTIVE POWERTRAINS FOR CONTROL STUDIES

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COMPUTER-AIDED MODELLING AND SIMULATION OF AUTOMOTIVE POWERTRAINS FOR CONTROL STUDIES

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INTRODUCTION

Many complex interactive processes exist in the design and development of new and advanced vehicle powertrains. These processes consist of the selection and matching of engine and transmission components, and the development of engine and powertrain controls which attempt to provide the optimum trade-off between fuel economy, performance and driveability, while adequately constraining the level of pollutants in exhaust gas emissions. Computer simulation is a valuable tool finding increasing application in the design of automotive powertrains, and in the analysis and control of the many interacting subsystems.

Advanced vehicle powertrains can conveniently be considered to consist of several interacting subsystems, viz. engine, transmission, vehicle, control system and driver, as illustrated in figure 1. In general, the individual subsystems are inherently complex, and result in an overall system which is multivariable, nonlinear and stiff (Jones et al (1)). In view of this inherent complexity, there is a clear need for the availability of sophisticated computer based analysis and design tools to aid in the formulation of powertrain control strategies and the subsequent optimisation of overall powertrain system performance. A minimum requirement is the availability of a simulation facility for exercising mathematical models of a particular powertrain of interest.

Many automotive powertrain simulations exist which are chiefly geared towards the improvement of fuel economy, either by better engine management, or better control and/or configuration of the powertrain. These can be sub-divided into those simulations which are specific to particular vehicles or vehicle configurations, and those which attempt to provide a general purpose tool for studying a wide range of vehicles. A further sub-division would separate the static from the dynamic performance simulations; and the causal from the non-causal, inverse-mode simulations.

This paper discusses the currently available approaches to the modelling of automotive powertrains and includes current work (1) in relation to the development of a Dynamic Automotive Powertrain Simulation (DAPS) facility. DAPS is a modular, user friendly simulation environment designed specifically for use in advanced powertrain control studies. The usefulness and flexibility of this facility is illustrated with simulation results relating to the control of an automotive continuously variable transmission (CVT).

APPROACHES TO POWERTRAIN MODELLING

To date the automotive control problems most frequently addressed relate to exhaust emissions and driveability-constrained optimisation of fuel consumption (Sweet (2)). Here, the performance metric for fuel consumption and emissions is based on a specific driving schedule, or schedules. Often the problem is treated simply as an internal-combustion-engine-control problem (Kuriger and Hughes (3)), which typically may result in microprocessor control of ignition timing, air-fuel ratio and exhaust-gas recirculation in an open loop manner (4). This approach usually involves a non-dynamic non-causal representation of a conventional powertrain and vehicle used solely to map the characteristics of the driving schedule to engine power requirements (Blumberg (5), Auiler et al (6), Bortz (7), Trella (8), Tennant et al (9), Matsumoto et al (10)).

Another sphere of problems is associated with the control of powertrains and hybrid vehicles. Again, as with engine calibration, the problem often posed is one of emission-constrained fuel economy; and is frequently addressed by simulations involving non-dynamic powertrains (Orshansky et al (11), Christenson et al (12), Radtke et al (13), Wallace et al (14), Baudoin (15)).

In order to develop an adequate representation of any particular system for simulation purposes, the scope of the simulation tasks needs to be specified. Without this knowledge each element or subsystem cannot be modelled with the appropriate level of detail, in order to describe the system efficiently. For instance, a transfer function model of a sub-system is useless if some of the internal variables need to be observed. Similarly it would be wasteful of effort and computing time to model, say, the motion of the engine valve train, if we wished only to observe the vehicle motion. The modelling itself is usually a combination of <u>a priori</u> knowledge, the application of physical principles, and the use of data obtained by experiment.

Engine models vary in detail from the static representations used in calibration studies ((6) - (15), Tennant et al (16)) to more detailed models combining physical principles with experimental data (Sherman and Blumberg (17), Hires and Overington (18), Morita et al (19), Morris et al (20), Bortz (21), Dobner (22), Powell (23), DeLosh et al (24)): these have application in areas such as the study of fuelling systems and the effect of torque transients on the powertrain. Most transmission and vehicle models used in fuel economy studies have been of the static type (5) rather than dynamic representations which incorporate such effects as tyre deformation and driveshaft twisting (24). The need for a driver model exists only in simulations which are causal, in that normal driver inputs, and some type of feedback is required in order to track a driving schedule ((24), Kuriger (25), Beachley and Frank (26)).

In the design and development of new and advanced vehicles, it is desirable to consider the automotive system as a whole. This is particularly important in control system synthesis and design, where a relatively small change in one parameter may cause considerable degradation of vehicle driveability due to unexpected subsystem interactions. The driveability problems associated with unwelcome interactions between the engine, the transmission and other elements of the powertrain are, to a large degree, concealed by the torque converter in an automatic transmission; and by the educated behaviour of the driver with a manual transmission. The torque converter allows smooth transmission from idle, and dampens the effect of abrupt changes in ratio. This has allowed the designer to treat the control of the transmission almost independently of the engine or vehicle dynamics.

The development of engine management and transmission control systems in almost total isolation from each other, does not really exploit the full potential for fuel economy and performance; and rarely does it yield a control that approaches the requirement for good driveability without considerable modification.

The development and use of continuously variable transmissions (CVT's) and energy storage devices, underlines the need for consideration of the dynamics of the total powertrain. The new configurations may accentuate the interactions present in the system, by removing, for instance, the decoupling effect of the torque converter. Also in such configurations the throttle may not be the best means of modulating the propulsive force. Evidently in order to address such control problems, simulation models and techniques must be used which allow the whole powertrain to be addressed as a dynamic system.

The versatility of the general purpose automotive simulations ((26), Waters (27), Hammond and Mcgehee (28), Bumby et al (29)) is their biggest attraction, allowing them to be used for a wide range of applications. The static performance simulations ((27) - (29)) neglect transient effects within the vehicle and powertrain components (other than inertia effects) therefore assuming that the vehicle and its powertrain move from one pseudo steady state condition to another. To include transient effects in detail requires a different modelling technique based on the numerical solution of a large number of differential equations. Another desirable feature, shared by HEAVY (28) and JANUS (29) is a high degree of modularity, allowing the rapid configuration of alternative systems. Neither of these programs is applicable to control applications and the study of transient phenomena, but they are both well suited for comparative concept evaluation, such as assessment of the benefits that may accrue for instance from the use of an electric motor instead of a conventional engine in a particular vehicle system.

The remainder of this paper discusses current work (1) relating to the development of a modular Dynamic Automotive Powertrain Simulation (DAPS) facility. DAPS is an interactive user-friendly simulation environment and represents the initial phase in the development of a comprehensive computer aided engineering (CAE) facility designed specifically for use in advanced powertrain control studies. DAPS has been developed in parallel with a control study related to a specific powertrain incorporating a continuously variable transmission (CVT). This control study has played a useful role in influencing the development of DAPS along lines appropriate to the needs of the automotive systems engineer.

SCOPE OF DYNAMIC AUTOMOTIVE POWERTRAIN SIMULATION PACKAGE

DAPS has been developed to equip the automotive systems engineer with a general purpose simulation tool for use in automotive powertrain control studies. The aim has been to provide a facility for creating and exercising dynamic automotive powertrain models, capable of being used by engineers with limited knowledge of computing and simulation whilst retaining the versatility required by experienced simulation engineers. The specific objective has been to design a computer simulation facility incorporating the following features:-

(i) a capability for efficient and accurate simulation of stiff dynamic systems.

(ii) an interactive mode of operation in which simulation commands can be issued one by one, and data and simulation results displayed graphically and modified on-line.

(iii) the ability to formulate a simulation model in terms of a high level language which provides a more convenient and direct form than conventional scientific languages such as FORTRAN, and allows for the model to be saved on file and analysed indefinitely.

(iv) the capacity for a modular approach to simulation model development, whereby distinct modules corresponding to the individual powertrain subsystems highlighted in figure 1 can be developed and tested independently.

(v) an automatic file management system for handling both the individual powertrain modules and the large quantities of data which can be generated during a simulation exercise, and providing a security system preventing unauthorised access to a particular user's simulation data and module files.

(vi) a user friendly man-machine interface based on menu driven dialogue.

(vii) a capability for expansion to include additional facilities together with a capacity for ease of enhancement of existing facilities.

Since the publication of the Continuous System Simulation Language (CSSL) report (30) in 1967, several general purpose simulation languages (Spriet and Vansteenkiste (31)) have been developed which meet the requirements (i) - (iii) outlined above, e.g. CSSL IV and ACSL (Advanced Continuous Simulation Language). ACSL (Mitchell and Gauthier (32)), in particular, is an interactive high level simulation language which operates on a fully time-shared basis (31). However, none of the currently available simulation languages (31), including ACSL, provides a dynamic simulation facility which, in addition to requirements (i) - (iii), fully meets the requirements (iv) - (vii) outlined above. This is not surprising since these simulation languages have been developed for general purpose usage and these latter requirements represent desirable attributes of a dynamic simulation facility designed specifically for use by automotive systems engineers in powertrain control studies. In view of this, DAPS represents a significant enhancement of the capabilities of existing high level simulation languages, tailored to the needs of the automotive systems engineer concerned with the control of advanced vehicle powertrains.

The DAPS facility ((1), Jones et al (33)) is based on ACSL and essentially provides an interface to ACSL incorporating the non-ACSL features (iv) - (vii). To meet these requirements, specifically in terms of flexibility of powertrain model definition and versatility in exercising it, the DAPS facility is split into:

(a) a Simulation environment,

and

(b) a collection of individual powertrain modules and simulation results

residing within this environment. In addition to providing an interface to ACSL, DAPS also provides a high level interface to the computer operating system.

The DAPS Environment is a set of interactive utility programs running under the control of a run-time executive. It provides for the automated management of powertrain modules and simulation results with a minimum of user effort, together with access to the comprehensive simulation facilities within ACSL. An important feature is its inbuilt security system which enables a user to define varying levels of access to his simulation results and powertrain modules and, in particular, to prevent unauthorised access to his modules or results. The executive and utility programs perform all complex computer related functions and, consequently, no knowledge of the computer file structure or operating system is required by the user. Use of the DAPS Environment is based on an interactive menu driven system incorporating a comprehensive HELP facility.

ILLUSTRATIVE APPLICATION: CVT POWERTRAIN

The development of DAPS has proceeded hand-in-hand with a parallel study concerning the mathematical modelling of an automotive powertrain incorporating a CVT. The CVT study has been specifically concerned with the development of a computer simulation model of the powertrain of the BL Technology experimental Perbury CVT Dolomite vehicle ((Curtis (34)) which incorporates a Perbury traction transmission (Curtis (35), Perry (36), Stubbs (37), Baker (38)) and an electronic control system (Ironside and Stubbs (39)) developed by the Lucas Research Centre (38). It is seen as a pilot modelling exercise aimed at identifying the experimental and conceptual background required for work on the control aspects of advanced automotive powertrains. In view of this, the scope of this pilot simulation model has been restricted to include phenomena associated with frequencies in the range 0 - 10 Hz. The model has been developed as a set of individual powertrain modules incorporated into DAPS and is described schematically in figure 2.

Experimental studies to obtain data such as moments of inertia, transient and steady state torque and speed relationships were pursued using facilities provided by the companies. This included the use of a test bed simulation rig at Lucas Research Centre and a hydraulic flow rig and the experimental Perbury CVT Dolomite vehicle, suitably instrumented, at BL Technology. The vehicle was also used to provide transient and steady state data for validation purposes.

Figures 3 and 4 describe the result of a simulation experiment performed on the powertrain model using DAPS and is intended to illustrate the usefulness and flexibility of the simulation approach. In this experiment the electronic control system module has been replaced by a simplified control system module which constrains the transmission to follow a ramp increase in ratio. This type of manoeuvre is difficult to achieve in a controlled manner in a vehicle but using the simulation model it is straightforward to quantify the effects of such a manoeuvre. This ability to subject a simulation model to simplified idealisations of actual manoeuvres is useful in control engineering terms since it can provide information to aid in the design of control systems. The results highlight the possibility that rapid changes in ratio might excite oscillation in the powertrain of CVT vehicles and is comparable to the response of a conventional vehicle following the rapid engagement of a clutch.

FUTURE DEVELOPMENT

DAPS represents the initial phase in the development of a comprehensive CAE facility for use in automotive powertrain control system design and essentially meets the minimum requirement in terms of the provision of computer based design tools tailored to the needs of the automotive systems engineer. An additional requirement is the availability of an interactive data analysis facility to aid in the processing and analysis of experimental data for incorporation into a powertrain model.

The experimental activities relevant to automotive powertrain control can range from the mapping of a complete engine on a test bed to the driving of an instrumented vehicle on a road or chassis dynamometer. Engine test bed experiments aimed at obtaining a static characterisation of an engine can, typically, generate 1 Mbyte of data whilst a single dynamic vehicle experiment might produce 150 Kbyte of data and, furthermore, a modest experimental programme with a vehicle would consist of ten or so experiments. The problems of managing such large quantities of experimental data are self evident and it is clear that there is also a need for a data base management system designed specifically for automotive systems applications.

Finally, a comprehensive CAE facility designed for use in automotive powertrain control system development should incorporate an interface to computer aided control system packages and, possibly, other computer aided design facilities.

CONCLUSIONS

This paper has discussed the currently available approaches to the modelling of automotive powertrains. It has concentrated on recent work relating to the development of a modular Dynamic Automotive Powertrain Simulation facility designed specifically for use in advanced powertrain control studies. DAPS represents the initial phase in the development of a comprehensive CAE facility for use in the design of of automotive powertrain control systems. Such a facility would also require the availability of an interactive data analysis facility together with a data base management system capable of handling the very large quantities of data associated with automotive powertrain experiments. In addition, such a facility should also be capable of interfacing with computer aided control system design packages and, possibly, other computer aided design facilities.

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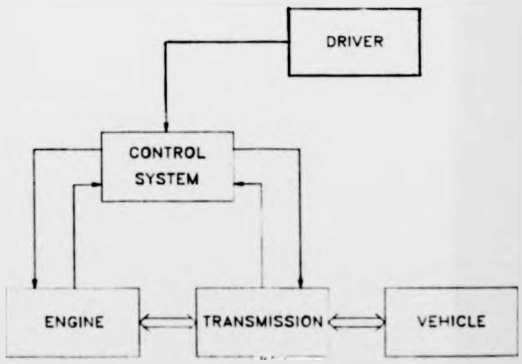

Figure 1: Advanced Automotive Powertrain

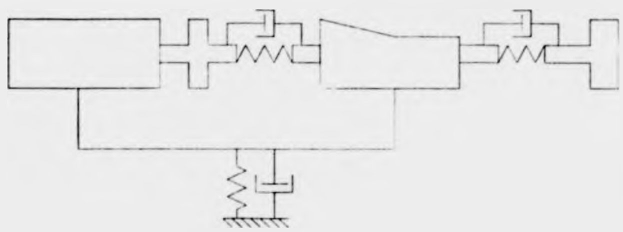

Figure 2: Schematic Representation of Model

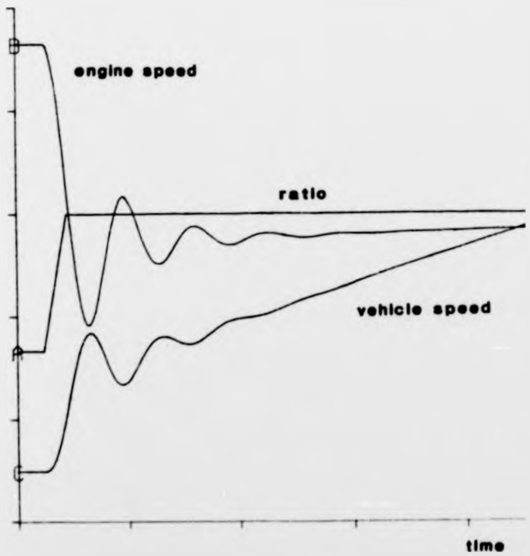

Figure 3: Response to Ramp Change in Ratio

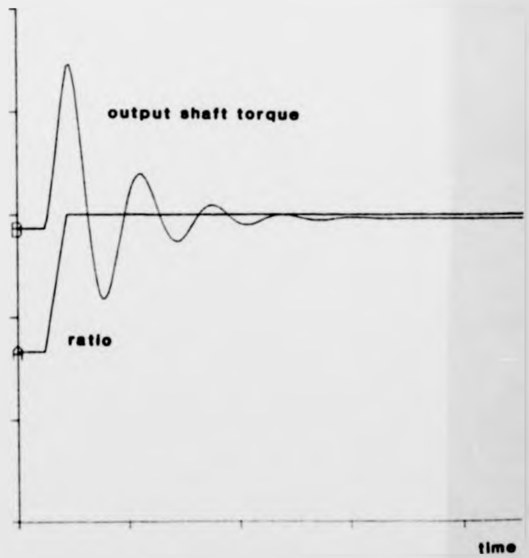

Figure 4: Response to Ramp Change in Ratio